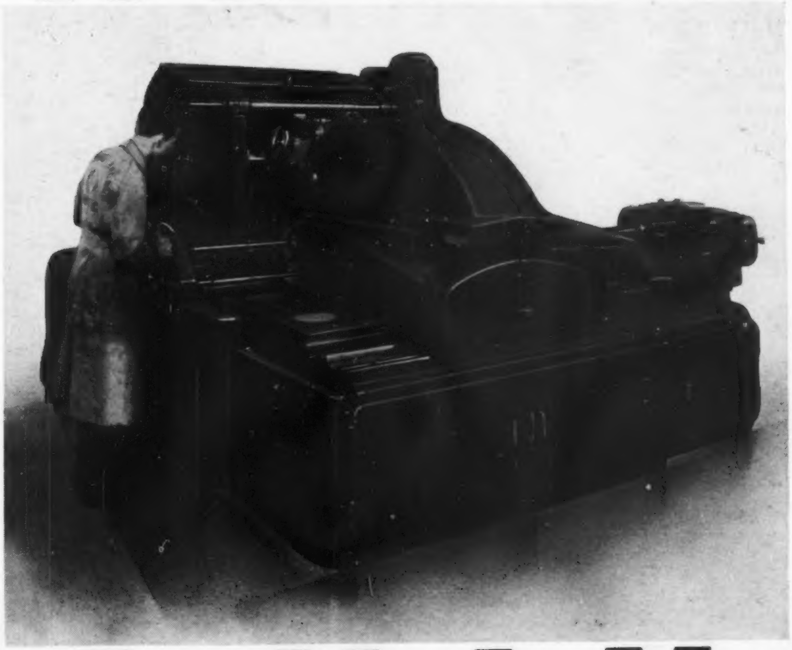
July 1941

MACHINE



DESIGN

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DEFENSE MACHINES DEMAND

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A 1416

ALLIS-CHALMERS LO-MAINTENANCE MOTORS

A COMPLETE LINE FOR EVERY PURPOSE...ONE-HALF HORSEPOWER AND UP

Topics

MECHANICAL jitters were recently removed from a turbine by Westinghouse research engineers through the simple expedient of welding pockets of loose sand onto the turbine housing. Friction between the grains of sand damps out all objectionable vibrations.

♦

FOR application as submarine cable insulation, a non-hygroscopic magnet core has been developed consisting of a pressed ferromagnetic powder and an insulating binder of polystyrene.

♦

DOMESTIC production of aluminum in the United States is at present about 600,000,000 pounds annually, nearly double that in 1939. By July 1942, production will have reached 825,000,000 pounds due to rapidly expanding facilities.

♦

NO LONGER will colored silk threads be used for markings in dollar bills. The imported threads will find a more than efficient substitute in Nylon fibers.

♦

RESOURCES of the General Electric Co. normally concerned with research and development of radio and television receivers will be devoted to vital defense production of electronic equipment for which there is an important need.

♦

NEW polarizing screen has been developed which is 99.99 per cent efficient. The screen is made of polyvinyl alcohol stretched from three to eight times its original length and then exposed to an iodine solution. The product transmits a third more light than the old type polaroid and is nearly free from color. Efficiency is so high that, when two sheets are crossed, a hundred watt lamp is not visible through them.

METHOD for obtaining built-in corrosion-resistant properties in tank ships utilizes composite plates of steel bonded together to form an electric potential claimed to resist corrosion. The plates reinforce one another and add structural strength to the ship without substantially increasing its weight.

♦

ALREADY in the blueprint stage are airplanes that will fly to Europe in 10 hours and others that will carry a full load of bombs to Europe or the Orient and return, according to Igor Sikorsky, engineering manager of Vought Sikorsky Aircraft Division, United Aircraft Corp.

♦

ESPECIALLY adapted for high temperatures is a new lubricant which does its work, then vanishes without a trace. Research engineers say it goes far toward solving the problems of lubrication in kiln cars, ceramics and glass molds, annealing and baking ovens, working parts of diecasting machines and various hot parts of machines in the metal industries.

♦

TO MEET pressing defense requirements, 15,000 employes of the United States Steel subsidiaries are now undergoing intensive training for skilled jobs—not only for the production of steel but also for the manufacture of defense items such as cruisers, destroyers, merchant vessels, shell forgings, and bomb casings.

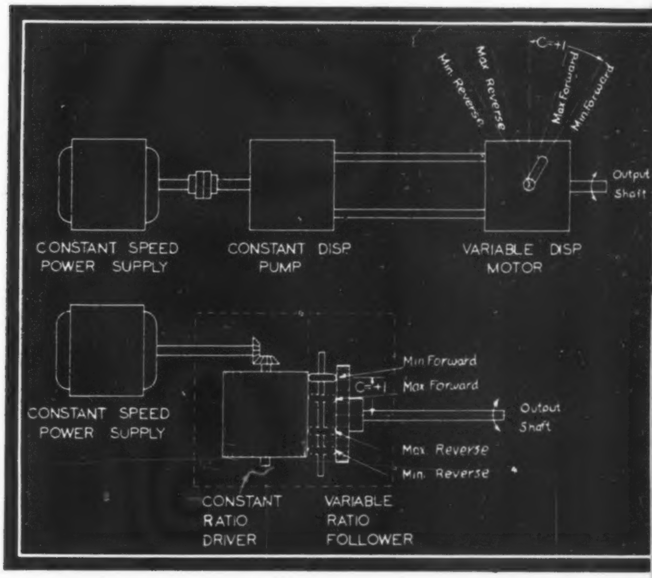
♦

MORE than 51,000,000 horsepower is required to turn the wheels of industry, according to the U. S. Bureau of Census. Purchased power from central stations accounts for 30,000,000 horsepower, the remainder being produced by prime movers in individual plants. Of the plants generating their own power, 16,000,000 horsepower is utilized for electric motor drives.

MACHINE DESIGN

Designing for Hydraulic Variable Speed

By Christian E. Grosser



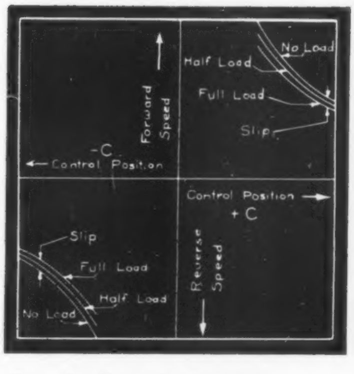


Fig. 1—Speed change is obtainable throughout range of forward and reverse by means of variable displacement motor. Characteristic curves for circuit are also shown.

COMMONEST among the applications of the hydraulic transmission is the driving of loads at variable speeds. In all fields of industry the demand for continuous speed variation in the driving of machinery is becoming increasingly insistent. For optimum production efficiency stepless adjustments to the speed of machine tool spindles and feeds are desirable. For smooth acceleration of high inertia loads without slipping clutches and for accurately controlled movement and positioning of large masses the variable speed hydraulic transmission finds its most advantageous use.

ANALYZING in detail the variable speed hydraulic circuit, this, the third of the series by Christian E. Grosser, is an extension of that portion of the December, 1940, article which dealt with speed variation. Subsequent articles will consider controlled torque, power and servo-circuits respectively

Most simply, the hydraulic variable speed transmission may be considered nearly the equivalent of an ideal change speed gearbox with an infinite number of adjustments;

the word "nearly" is introduced as qualification since the hydraulic transmission is subject to a small amount of slip due to leakage. In this respect it might be said that hydraulic control of speed is not positive. However, slip is usually so small that its effect is negligible. The nature of this slip and its significance in certain drive applications will be taken up in some detail later.

Broadly speaking, controlled variable speed hydraulic transmission circuits such as that used in the planer, *Fig.* 3, may be classified under three main groups according to the manner in which the drive speed responds to the external control:

1. Output or motor shaft speed is varied by a changeable pump displacement which directs a variable flow rate of liquid under pressure to and through a constant displacement motor. The hydraulic motor shaft speed is dependent almost directly on the volume rate of flow passing through the motor, *Fig.* 1
2. Output speed is varied by a changeable hydraulic motor displacement. The motor is supplied with a substantially constant flow rate of liquid under pressure from a constant displacement pump, *Fig.* 2
3. Output speed is varied by changeable displacements of both pump and motor, *Fig.* 4.

It is implied in all the illustrations of hydraulic speed transmissions under discussion that the input power to the transmission (i.e. at the pump shaft) is supplied at a substantially constant speed, as for example from a single-speed induction motor.

Clarified by Mechanical Analogy

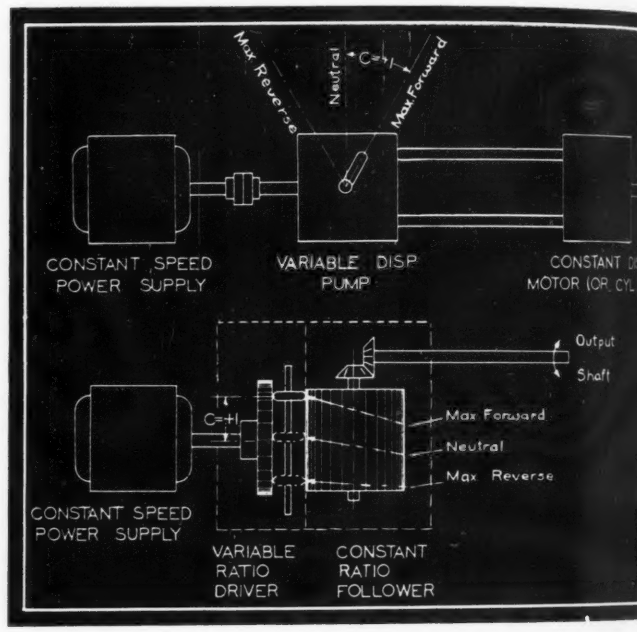
Each of the three foregoing circuit types has distinct characteristics with respect to capacity to transmit power over the variable speed range, the available range over which output speeds may be varied, and the response of the output speed to the control position. These characteristics are shown graphically in *Figs.* 1, 2 and 4. Also the kinematical equivalents are shown in the forms of schematic mechanical friction drive combinations.

Relative capacity of each circuit to transmit power at various speeds may be perceived by studying the friction drive combinations in each case. If it is assumed that the contact pressure between the friction rolls is limited to some maximum value, we note that in *Fig.* 1 the output power capacity in a given size unit varies directly as the output speed since the maximum output torque is restricted to a constant value dependent upon the size of the follower roll and the pressure of contact. Similarly in the hydraulic transmission shown in *Fig.* 1 the maximum output torque capacity is limited to a constant value dependent upon the displacement of the hydraulic motor and upon the maximum permissible liquid pressure in the circuit.

In *Fig.* 2 it may be noted that the output torque capacity under given roll contact pressure is in inverse ratio to the output speed. This means that the arrangement has the capacity to transmit a constant amount of horsepower at any speed within its range of speed control. The same property of constant power is had by the hydraulic circuit.

Fig. 4 shows a combination of the types in *Figs.* 1 and 2. The power-carrying capacity of this arrangement is constant for a given setting of the pump control, and proportional to the displacement.

Circuit shown in *Fig.* 1 is most commonly used.



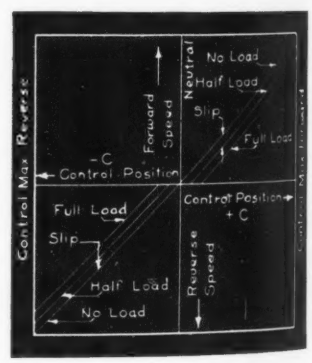


Fig. 2 — Differing from Fig. 1 in that a variable displacement pump is used, mechanical analogy and characteristic curves provide a comparison of the two systems

In place of the variable displacement pump the smaller size transmissions often employ a constant displacement pump with a flow control valve in series, for reasons of lower cost and simplicity.

Circuit in *Fig.* 2 is often used in applications where a reduction in speed requires a proportionately increased torque. The practical speed range is usually restricted to 2:1 or 3:1, and obviously it cannot be used for variation of speeds through zero or for reversal.

Circuit in *Fig.* 4 is the general form, not in extensive use, but shown here for its value to analysis since it combines in one arrangement the properties of the other two circuits.

It is interesting to note that the "slip" or loss in

speed occasioned by leakage in a hydraulic system is well illustrated, in the friction drive equivalent, by the slip of the rollers at their contacts. The speed variation for the mechanical arrangement shown in *Fig.* 4 may be represented by:

$$\pi N_f D_f C_f = \pi N_d D_d C_d - S \quad \ldots\ldots (1)$$

Follower roll surface speed	Driver roll surface speed	Total roller slip

where

N = shaft speed in revolutions per unit of time
D = maximum diameter of drive roller
C = fraction of full drive roll diameter at point of operation. Maximum control positions represented by +1 or —1
S = total speed of slip at roller contacts
f = follower
d = driver.

By transposing Equation 1 becomes:

$$N_f = \frac{D_d C_d}{D_f C_f}\left[1 - \frac{S}{\pi D_d C_d N_d}\right] N_d \quad \ldots\ldots\ldots\ldots (2)$$

where the factor in brackets might be called the "speed" efficiency of the drive. The diagram at the right of *Fig.* 3 illustrates this function.

In the hydraulic transmission the same sort of relationship holds. If we consider the working liquid to be incompressible, continuity of flow prescribes:

$$N_m D_m C_m = N_p D_p C_p - Q_L \quad \ldots\ldots\ \ldots\ldots (3)$$

Motor displacement rate	Pump displacement rate	Total volume rate of leakage

N = shaft speed in revolutions per unit time
D = maximum volumetric displacement per shaft revolution
C = displacement control setting as fraction of full displacement. (Full disment = 1)
Q_L = volume rate of total leakage from system
p = pump
m = motor.

By transposition (3) becomes,

$$N_m = \frac{D_p C_p}{D_m C_m}\left[1 - \frac{Q_L}{D_p C_p N_p}\right] N_p \quad \ldots\ldots\ldots (4)$$

where the factor in brackets may be called the volumetric efficiency of the transmission in terms of the pump displacement rate. Equation 4 is also represented by the diagram beneath *Fig.* 4.

It may now be seen that circuit 1 and 2 are only special cases, or simplifications, of 3, since in each the constant displacement element prescribes the constant value of unity for its control factor. In circuit 1 Equation 4 applies if $C_m = 1$. In circuit 2 the equation holds if C_p is considered as being equal to unity.

Examination of Equation 4 shows that if the volumetric efficiency is 100 per cent—that is, there is no leakage—the ratio of output to input speeds is determined by the factor D_pC_p/D_mC_m. By selection of the fixed ratio D_p/D_m it is possible to introduce a basic reduction ratio into the transmission which will be the ratio of the maximum output speed to the constant input shaft speed. For example, a 10 cubic inch per revolution pump operating at 1200 revolutions per minute would drive a 20 cubic inch per revolution motor at a maximum speed of 600 revolutions per minute. The variable factor C_p/C_m is then the control function which varies the output shaft speed from zero to its maximum in either direction.

For optimum design it is best to maintain the factor $D_p/D_m = 1$, that is to say, pump and motor should be of equal size. This is for the reason that with unequal size units operating together, either the pump or the motor must operate at less than its rated maximum speed. Examination of the volumetric efficiency factor (considering leakage dependent only on the unit sizes and pressure, since it is substantially unaffected by speed of either pump or motor) shows that it is highest when the pump and motor are smallest and operating at their highest possible speeds.

Leakage in practically all forms of hydraulic apparatus is almost directly proportional to the

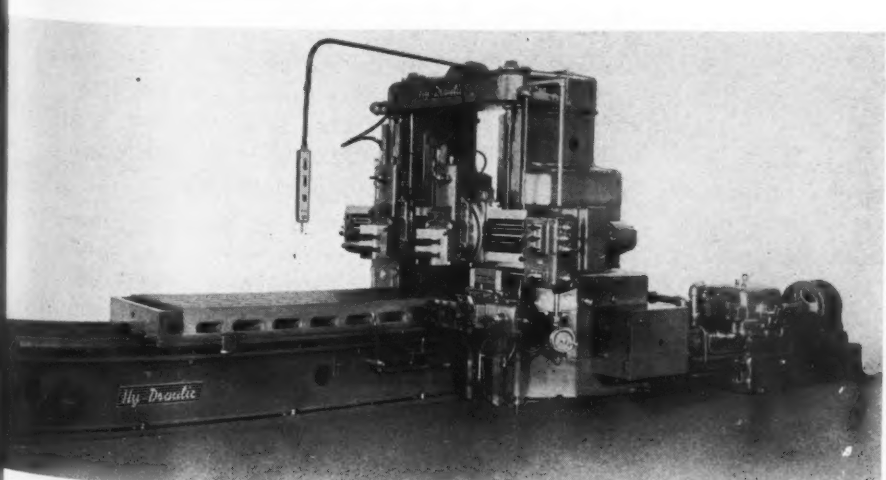
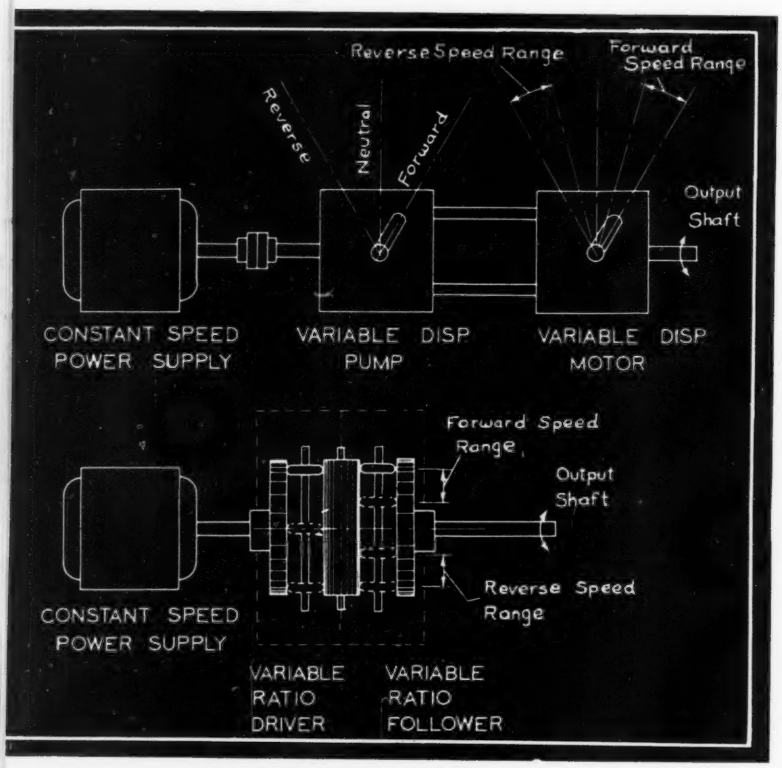


Fig. 3—Hydraulic drive and feeds on Rockford planer assure smooth, positive control from constant speed electric motor

liquid pressure. This is because clearances are small and the viscosity of ordinary working fluids at operating temperatures is such that leakage is in laminar flow for all except extraordinarily large clearances. As a consequence, leakage is dependent only upon the load resistance, since the load prescribes the liquid pressure. At rated maximum loads hydraulic transmissions consisting of a pump and motor of equal size and operating at maximum speed will usually show volumetric efficiencies in



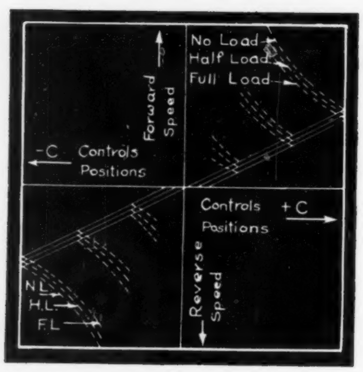


Fig. 4—Combination of circuits shown in Figs. 1 and 2. With similar hydraulic equipment, a wider range of adjustment is possible

the range .90 to .95. Expressed differently, a well designed transmission will leak from 5 to 10 per cent of its full pump displacement rate under its maximum rated pressure. For more accurate prediction of the volumetric efficiency of a transmission and the consequent effect of operating pressure on the output speed, it is best to find the leakage rate of each element in the circuit at the operating pressure. Summation of the individual leakages at a given pressure yields Q_L which may be substituted in Equation 4 to find the operating output speed at that pressure. Most commercial pump and hydraulic motor catalog data will include leakage or volumetric efficiency figures. For other sources of leakage, such as in control valves and pistons, the viscous flow formula may be applied:

$$Q_L = \frac{.0378(p_1 - p_2)C^3 w}{Dl}$$

where

Q_L = leakage flow rate in cubic inches per minute
$(p_1 - p_2)$ = difference in pressures at ends of clearance in pounds per square inch
C = clearance in thousandths of an inch
w = clearance width in inches
l = clearance length in inches
D = oil kinematic viscosity in centistokes

Viscosity data for various oil temperatures is obtainable from oil manufacturers. Common transmission oils lie in the range SAE 10 to SAE 20. Operating temperatures are usually in the neighborhood of 150 degrees Fahr., at which temperature the kinematic viscosity of SAE 10 is approximately 15 centistokes; for SAE 20 it is 20 centistokes.

Maximum speed ratings for almost all types of positive displacement pumps and motors may be taken to follow the general rule:

$$RPM_{max.} = \frac{K}{\sqrt[3]{D}} \qquad (5)$$

where D = displacement per revolution in cubic inches, and K = design constant which has values close to 2000 for most transmission type hydraulic pump and motor units.

The relationship, Equation 5 is rational inasmuch as it maintains at a constant value the dynamic stresses due to rotation for all sizes of a given design of unit. The rule also maintains a constant velocity of fluid flow through a size series of a given type of unit. The value of K will be found by examination of manufacturers' speed ratings to vary over a wide range. This variation is due in some instances to a desire to standardize on ordinary alternating current induction motor speeds for pump drives. One size of a given design of unit will generally be found to be rated at a value of K at least 2000 or larger. Unless there are special circumstances governing the design of the other sizes, they may logically be operated at speeds determined by this same value of K. However, it is best to consult the manufacturer in all cases before departing from published speed ratings.

Capacity of a transmission to transmit power involves the torque ratings, as well as the speed ratings, of the pump and motor in a circuit since power is proportional to the product of torque and speed. Torque capacity depends in turn upon the maximum pressure which the circuit components will withstand. From examination of *Fig.* 5, which represents the most simple type of rotary shaft pump, the theoretical relationship is derived:

$$T = \frac{D \Delta p}{2\pi} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(6)}$$

where T = shaft torque in inch pounds, D = pump or motor displacement per revolution in cubic inches, and Δp = difference in the pressures at the inlet and outlet of the unit in lbs./sq. in.

Due to internal mechanical and fluid friction losses, the actual shaft torques will depart somewhat from the theoretical value given by Equation 6. The actual values may be obtained by the expressions:

$$T_p = \frac{D_p \Delta P_p}{2\pi} \times \frac{1}{y_{Tp}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \text{(7)}$$

$$T_m = \frac{D_m \Delta p_m}{2\pi} \times y_{Tm} \quad \ldots\ldots\ldots\ldots\ldots\ldots \text{(8)}$$

where y_T = torque efficiency.

The factor y_T for pumps and motors depends upon both load (or pressure) and speed. Fluid friction rises with increase of speed. Also no-load losses due to packing friction, etc., play an increasingly important part under low operating loads. It is practically impossible to define y_T accurately for all conditions of operation. However, it will be found that over a large part of the operating range near rated maximum pressure and rated maximum speeds the torque efficiencies of pumps and motors will be close to .95. This value may be used in a first approximation design and checked later with the actual characteristic data of the selected units. The difference between this assumed and actual values usually will not affect a design materially.

Piping Introduces Losses

It will be noted that the maximum pressure, $p_{max.}$, imposed upon the circuit will occur at the discharge of the pump, since there is a continual fall of pressure as the liquid flows through the piping, motor, and back to the intake of the pump. The value of Δp_p, in Equation 7, which is the pressure drop across the terminals of the pump, may be taken equal to the maximum rated pressure of the system since the pump intake pressure is nearly atmospheric. If supercharging pressure is required at the pump intake this should be subtracted, from p_{max} but its omission is not serious since supercharging pressures are usually less than 5 per cent of the maximum system pressure. The value of Δp_m in Equation 8 will always be equal to the pressure drop across the pump less the total pressure drop in the connecting piping circuit. This is

$$\Delta p_m = \Delta p_p \left(1 - \frac{\Delta p_{CL}}{\Delta p_p}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots \text{(9)}$$

where Δp_{CL} = pressure drop along all piping, in lbs./sq. in.

Maximum system pressures vary from 500 to 3000 lbs./sq. in. Most common pressures are near 1000 lbs./sq. in. The value of Δp_{CL} may be controlled by providing sufficiently large connecting lines, avoiding sharp corners, sudden changes of pipe section, etc. By using in reasonable lengths, the pipe sizes recommended by the manufacturers of the circuit components the value of $1 - \Delta p_{CL}/\Delta p_p$ may generally be kept above .95. An empirical rule for determining proper pipe sizes in ordinary size transmission circuits is to restrict the velocity of flow to 20 feet per second. A better method is to check the actual pressure drop through all the connecting circuit by pipe friction formulas and select such a pipe diameter as will restrict this quantity to approximately .95 which may, for convenience, be considered the efficiency of the connecting circuit.

As a concrete illustration of the use of the design considerations stated in the foregoing it may be of value to set down a definite example of a transmission design to satisfy typical conditions.

Considering the drive of a conveyor which is to operate against a substantially constant torque load

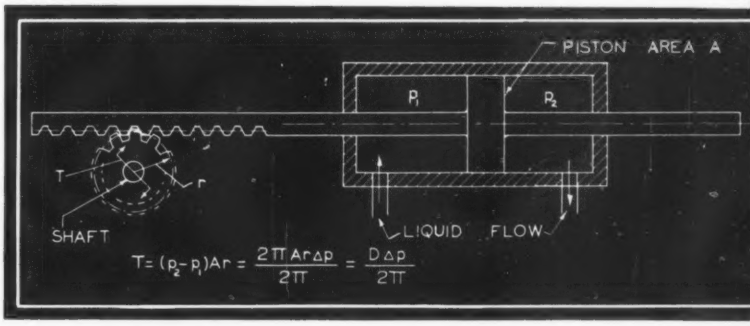


Fig. 5—Diagrammatic sketch illustrates principle of rotary displacement pump and facilitates derivation of torque relationship

in a speed range from 0 to such a maximum speed that the maximum input power is 10 horsepower, we note the following:

1. Circuit 1 is adaptable since the maximum torque load at all speeds is essentially constant
2. Under the assumption that the same size pump and motor are to be used and since the overall efficiency of the transmission is the product of the component efficiencies, the overall efficiency at maximum load may be conservatively estimated as

$$y_{overall} = y_{Tp} \times y_{Tm} \times y_{CL} \times y_V \quad \ldots\ldots\ldots\ldots \text{(10)}$$
$$= .95 \times .95 \times .95 \times .90 = .77$$

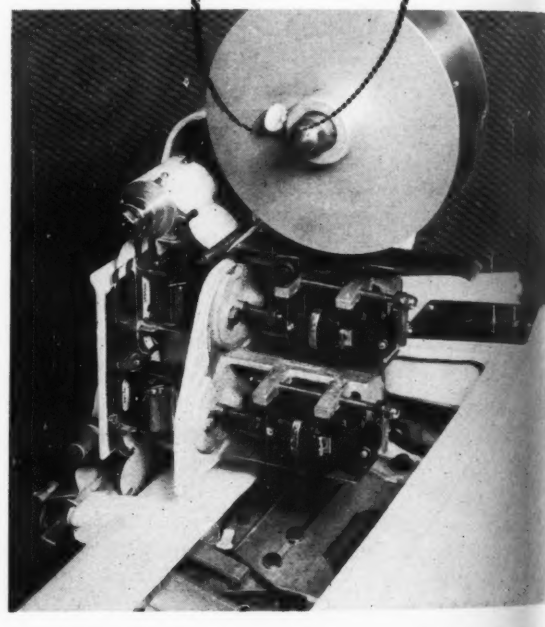
3. Gearing will probably be necessary between the transmission output shaft and the conveyor shaft, with efficiency of perhaps .95
4. The driving electric motor will therefore need to be

$$HP_{motor} = \frac{10}{.77 \times .95} = 13.7 \quad \ldots\ldots\ldots\ldots \text{(11)}$$

for which a 15-horsepower motor will be suitable
5. By the combined use of Equations 5 and 8 we may obtain the minimum required displacement of the equally sized pump and hydraulic motor, under the assumption that

(Concluded on Page 116)

Scanning the Field FOR IDEAS

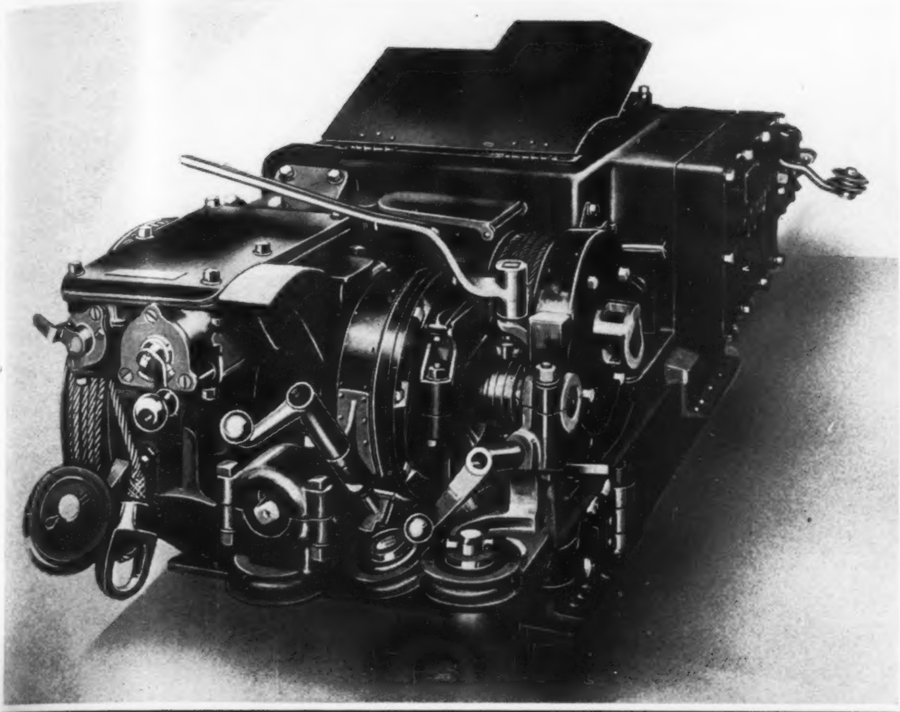


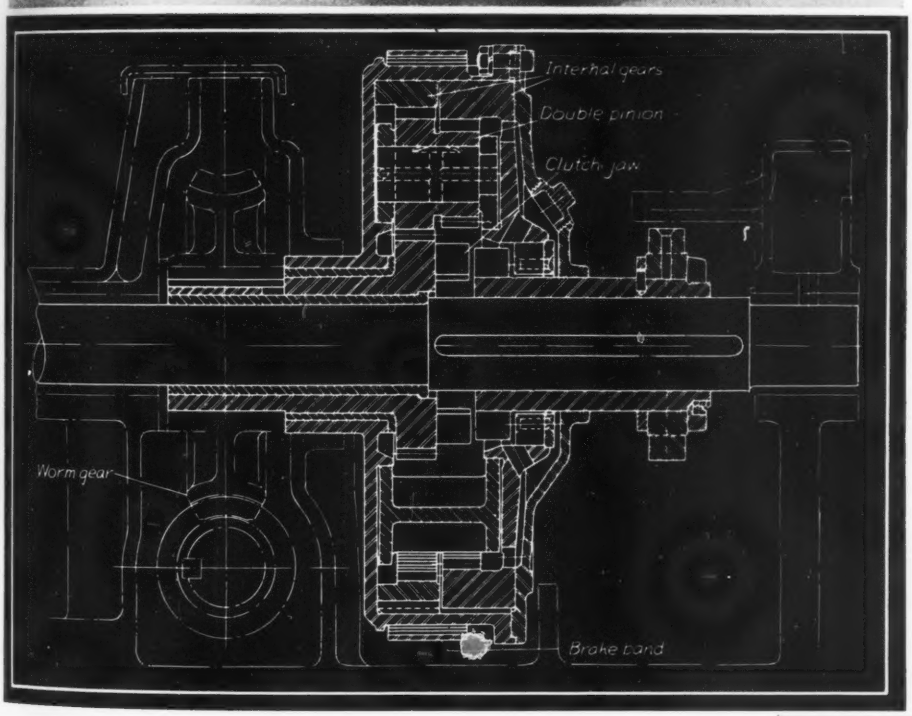
Positive, chain-like feed on this roll type labeler is provided simply by the labels themselves. Lugs on the driving drum, as shown in the illustration at the left together with a similar chain driven drum, index the labels for coding operations and assure that a full label is properly affixed to the package being marked.

Coding device consists of two simple solenoid-operated mechanisms as shown. This machine, designed by the Oliver Machinery Co., is equipped to stamp the day of week and code numbers in one of three manners: Consecutive, duplicate and repeat.

Magnetization of steel tape on this instrument, below, developed by the Heller Magnograph Corp., records simultaneously from a number of different sources so that desired phenomena may be synchronously recorded, co-ordinated and analyzed. Any data capable of being transformed into electrical impulses can be transcribed on this instrument. Stress, strain or vibration in aircraft as well as readings of any of the control instruments are typical.

Eight separate channels can be recorded on one tape, simultaneously, consecutively or in any combination. The tone head acts both as a recorder and a playback. Also when tape is desired for reuse, the same head is employed as an eraser. The head consists of eight sets of coils and cores spaced transversely so that no cross interference between channels is experienced. Variations in impulses to the coils cause corresponding degrees of magnetization in the proper tape channel.



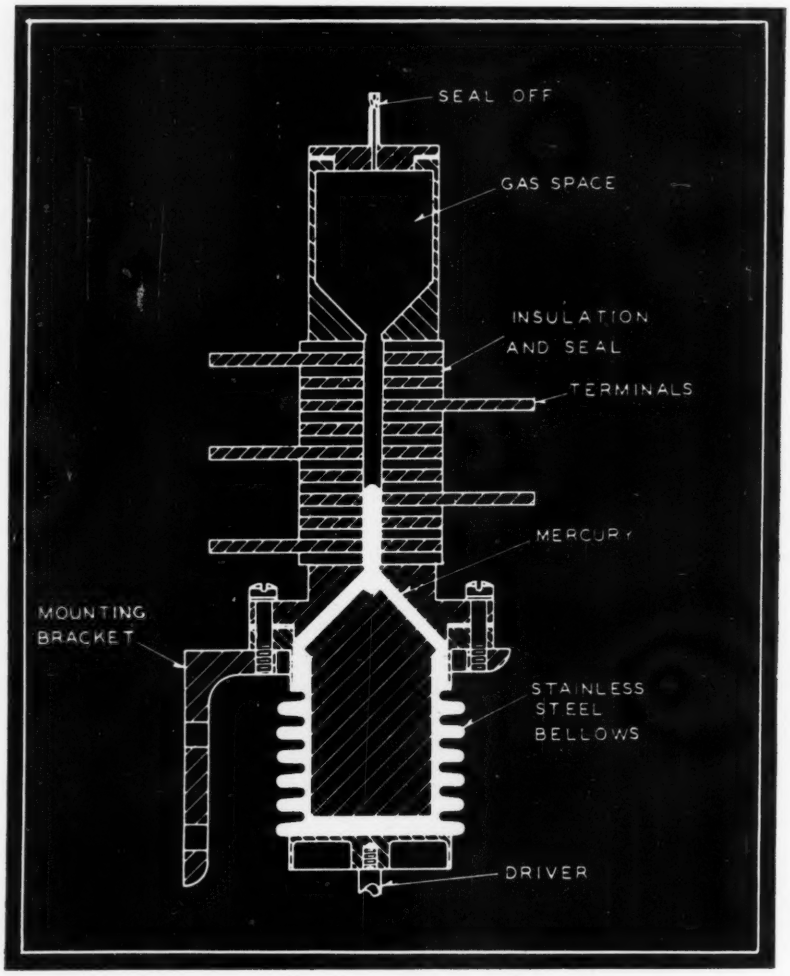


Overload protection is obtained in the two-speed, compact drive illustrated in the photograph and drawing at left. Designed by Goodman Manufacturing Co., the power transmission drives either direct or through a planetary for its speed change. When driving through the latter a brake band can be adjusted to slip at the desired overload.

Operation is as follows: Worm gear is attached to the sun gear of the planetary which has a set of clutch jaws. One of the internal gears is keyed to the inside of the brake drum while the other has another set of jaws. When the sun gear jaws are engaged, there is a positive direct drive. When, however, the sliding jaws engage the internal gear, the planetary is in operation. If the brake band is tightened, one of the internal gears is stationary while the other is rotated slowly by the differential diameters of the planet pinions.

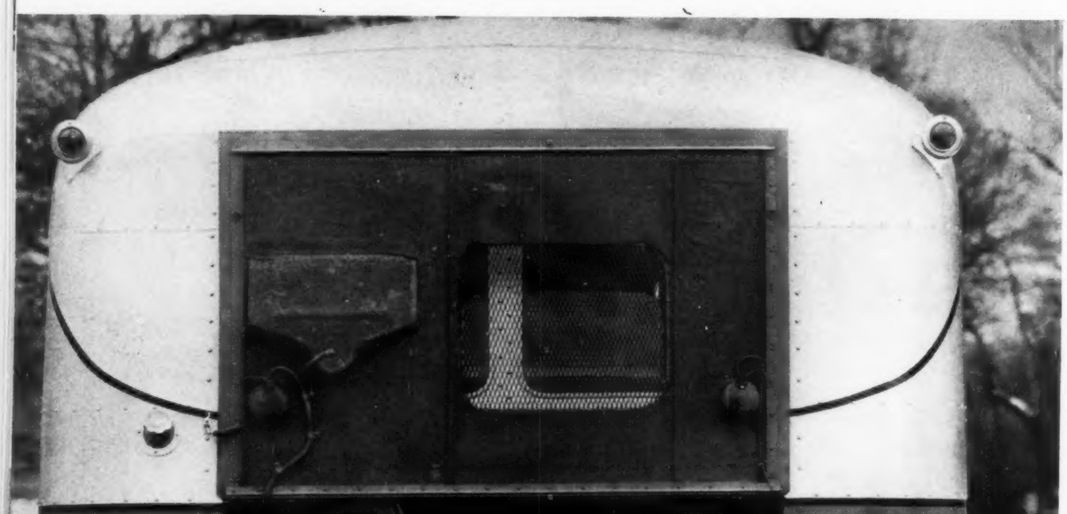
Visual signals aid in the operation of many machines to increase their production and ease of control. Recognizing this, the designers of the electric typewriter at the right have utilized a red and green lamp unit that not only signals at the end of a line like the conventional bell but also gives a warning when the end of a sheet of paper is reached. Thus a typist is able to give more attention to the transcribing at hand.

Signal for end-of-paper is simply composed of a narrow groove on right side of platen and a narrow finger controlling a switch. After the end of the paper passes, the finger enters the groove and closes the signal circuit.



Volume differential of the type that reminds one of the traditional mercury thermometer is utilized in the 40-contact mercury switch at left. In this way a few thousandths of an inch displacement of a bellows results in a two-inch movement of the column of mercury, shorting all the contacts. Designed by the Westinghouse Research Laboratories the switch is for use where only small amounts of power are available for operation. Hysteresis resulting from frictional and magnetic forces is necessarily small, as is the inertia of moving parts.

Power capacity is limited by the current interrupting ability of the contacts. The design illustrated easily can handle six kilowatts. With an initiating force of .05 watt to operate the bellows, a power amplification of 120,000 is effected.



Vibration control is usually an important consideration in the design of a machine. In the Twin Coach bus at left the engine, mounted at right angles to the drive shaft, is cushioned on eight rubber mountings of the shear type. Three-point cushioning is provided with four-point stability. Vertical snubbing features of these Lord mountings protect the engine from road shock by using metal snubbing washers on the top and bottom of each mounting. The resulting movement is a slight deflection of a center pin in the mounting which is sufficient to bring the correct area of rubber into contact with the metal washer. Motion due to road shock is only a few thousandths of an inch.

Fig. 1—Radial drill utilizes aluminum bronze for highly stressed gear bearings and nuts

Aluminum Bronze Fills Design Need

By J. D. Zaiser
Asst. General Manager
Ampco Metal, Inc.

ADVANTAGES accruing from the use of aluminum bronze in machine applications were known prior to 1910. However, these benefits were not fully realized due to production difficulties which resulted in wide variations in machining, casting and service properties. These difficulties have been overcome and, under present conditions, aluminum bronze alloys are rapidly assuming their rightful place among materials available to the designer of machines.

There is no single outstanding property of aluminum bronze which serves as a yardstick for its application. Certain basic characteristics common to the series of alloys, however, might be classified as follows:

1. Uniformly good physical properties. These can be varied with the alloy composition and heat treatment, but all resulting alloys will have tensile strengths in excess of 60,000 pounds per square inch and yield strengths in excess of 20,000
2. Controllable hardness range. The hardness is not a fixed quantity, but can be controlled within close limits through its range by varying the alloy composition and by heat treatment
3. Good corrosion resistance
4. Retention of physical properties at high temperatures. In those instances where it is necessary or desirable to use bronze at temperatures up to 1000 degrees Fahr., aluminum bronze is a practical material
5. High impact strength and high endurance limits. Izod values for aluminum bronze are higher than for any other bronze alloy of over 60,000 pounds per square inch, tensile strength.

In addition to these points it might be well to note that, with few exceptions, aluminum bronzes can be heat treated for the improvement of specific properties, can be forged, rolled, extruded, and spun.

Included in the category of bearing applications are not only bushings but also slides, gibs, thrust bearings, and similar types of parts. This is a large field of service for aluminum bronze, but one in which the alloy cannot be indiscriminately applied. Aluminum bronzes, since they contain no lead, are not possessed of so-called "self-lubricating properties" which will enable them to serve under conditions of little or no lubrication. Actually the only bronzes which are truly self-lubricating are those which are oil or graphite impregnated, but leaded bronzes are also widely used for poorly lubricated service with comparative satisfaction. Aluminum bronzes as a class are hard enough and strong

***CONSTITUTING** the third of MACHINE DESIGN'S series on engineering materials, this article deals with physical properties and machine applications of aluminum bronze. Its use as a bearing material, for gears, in corrosive service, etc., receives detailed treatment invaluable to the designer*

TABLE I

Physical Properties of Aluminum Bronze

No.	Tensile Strength (1000 lb./sq. in.)	Yield Strength (1000 lb./sq. in.)	Elongation % in 2 in.	Brinell (3000 kg.)	Composition (per cent)	Application
1.	70-77	25-32	30-40	115-126	Al 9-9.5 Fe 1.25 max. Cu balance	As cast, low-iron type. Good corrosion resistance. Also recommended for installations subject to shock or impact; low brinell prevents use for heavy loads.
2.	80-90	40-50	22-30	143-156	Same as No. 1	Heat treated alpha alloy for bearing and gear service. At sacrifice of some of the extremely good elongation of alloy No. 1, other properties are improved.
3.	80-100	40-50	12-15	163-179	Al 9.5-10 Fe 1.25 max. Cu balance	Heat treated alpha-eutectoid alloy; particularly suited to heavy loading and high stress.
4.	75-85	35-42	12-18	146-166	Al 10-12 Fe 2.5-4 Cu balance	As cast, high-iron type with good corrosion resistance. Also recommended for mechanical installation under heavy unit loading, or subject to shock. Endurance limit, 35,000 lb./sq. in.
5.	90-105	43-50	8-12	180-200	Same as No. 4	Heat treated variation of alloy No. 4. High brinell hardness coupled with good elongation and high strength. Widely used in machine tool and aircraft industries. Has good resistance to wear and abrasion.
6.	70-80	42 min.	1-4	285-311	Al 13 Fe 4.5 Cu balance	As cast high-iron, high aluminum bronze; used primarily to replace hardened steel for forming and drawing dies, wear strips, guide bushings.

enough to seize and damage the shaft or mating steel part in the event of oil failure.

One point that should be borne in mind is that proper bearing installation for aluminum bronze will go far toward insuring trouble-free service. The part should be accorded a good finish on the working surface because aluminum bronze will not "wear in" as freely and rapidly as softer materials. Once the initial "running in" period has passed, no trouble need be anticipated.

Aluminum bronzes containing over 10 per cent aluminum conform in structure with the usually accepted theory of a good bearing material; namely, hard particles supported in a softer matrix. In addition, certain of these alloys contain an effective concentration of a supplementary structure which consists of very small particles of an intermetallic compound, finely divided. As nearly as can be determined, these particles provide an added measure of resistance to wear with increased life.

Versatility of aluminum bronzes in bearing services is illustrated by the fact that press bearings of a soft aluminum bronze (126 brinell hardness number) have successfully replaced babbitt and afforded greatly increased life; while at the other end of the scale, two builders of precision boring machinery use a hard aluminum bronze (300 brinell hardness number) in place of the hardened steel previously used, for boring-bar guide bushings—an installation where long-lived accuracy and freedom from wear is essential.

While it may be largely true that aluminum bronze is best suited for comparatively slow speed applications, it has been used with entire satisfaction for spindle bearings on precision surface grinders where surface speeds run as high as 1200 feet per minute with bearing clearance of only .001-inch on a 3-inch diameter. This is a specific in-

Fig. 2—Typical machine parts of aluminum bronze include clutch disks, shifter forks, gears, bearings and slides

Fig. 3—Boom hoist worm gear used in earth moving machinery is representative of the heavy duty service to which this material is applicable

stance of the value of proper bearing preparation, for had these bearings not been carefully finished before installation, seizure would have resulted.

High compressive strength of aluminum bronze and its high impact strength particularly suit it for heavy duty service. In the steel industry, for example, aluminum bronze (brinell 160-180) is widely used for the universal joint bearing segments between rolls and mill drive. Here, under conditions of terrific pounding and the heavy loading that necessarily accompany the reduction of steel blooms into sheet, aluminum bronze performs satisfactorily. Thrust washers or pads on top of the upper roll bearing of the steel mill, against which the roll screw bears, are aluminum bronze, as are the screwdown nuts which control roll spacing adjustment and which must absorb the pressures that accompany the reduction of steel blooms, billets or sheet.

Earthmoving and construction equipment machinery manufacturers also find that aluminum bronze (brinell 160-180) assures trouble-free operation for track roller bushings, turntable roller bushings and, in the case of dredges, cutter head bushings. In both this and steel mill service a factor which may well cause premature failure is the presence of abrasive particles of sand, dirt or mill scale. It has been found that due to the hardness of aluminum bronzes these particles will not imbed themselves and score mating parts but are rejected and carried off with waste lubricant.

In addition to heavy machinery application, aluminum bronze has been widely adopted by machine tool and engine builders for certain parts such as bushings, cams, cam rollers, slides, shifter forks, feed fingers, and chucking shoes, where softer bronzes have been found to fail either through wear, deformation or plastic flow under high unit loadings. Radial drill, *Fig.* 1, is outstanding for its several bearing applications of aluminum bronze.

Certain unfavorable factors relating to coefficient of friction, machinability and, in some cases, higher costs militate against the wider use of aluminum bronze for bearing service. However, since these same factors have effect on other applications, they will be discussed later in this article.

Has Low Friction Coefficient

It is generally conceded that, due to the lower frictional coefficient of nickel bronze, it is the more suitable material for worm gears operating at extremely high speeds coupled with light to moderately heavy loading. Aluminum bronze under such conditions, unless carefully selected, may have a tendency to overheat and run at temperatures in excess of the allowable maximum.

Applications in such machine parts as are illustrated in *Fig.* 2, wherein aluminum bronzes have proved outstandingly satisfactory, are those in which loading is extremely heavy, the installation is subject to impact loading, operation is intermittent, or a combination of all these. As in many other types of mechanical service, however, there are many borderline cases where service life difference is not a matter of days versus months, and where nickel bronzes, tin bronzes, manganese bronze, and aluminum bronzes will all serve long enough to pass the minimum requirements of successful performance. In such instances it is up to the engineer to decide whether the higher yield strength, impact value, hardness, and longer life of aluminum bronze parts such as the worm gear, *Fig.* 3, is of enough value to him and to his machine to justify a price which may be a few cents per pound higher than other bronze alloys.

Widest users of aluminum bronzes for gears are heavy machinery builders and machine tool manufacturers; the former because they require highest possible physical properties in tension, shear and compression, the latter because of the material's wear resistance and good proportional limit.

Alloys Resist Corrosion

The alloys have one outstanding physical attribute which probably accounts, more than any other, for their excellent general corrosion resistance—the almost immediate formation of a film of aluminum oxide on the alloy's surface, whether cast or machined. That film, which although thin is closely adherent, has high resistance to attack by the majority of acids and some alkalis, although it is not equally effective with all corrosive media. Further, the film is self-healing; that is, when re-

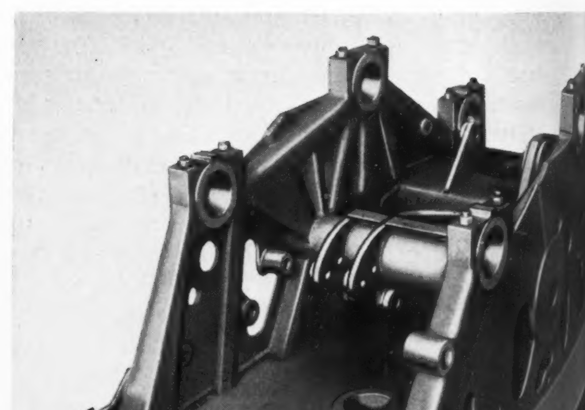
(Continued on Page 118)

Among the advantages of welding main frames and mechanical parts of a machine are lightness in weight, fewer parts, relatively homogeneous joints, and various kinds of metals combined into a single part. The P & H excavator illustrated is an excellent example of the extent to which welding can be applied in machine construction to achieve these advantages and is especially interesting because of the current interest in track-laying vehicles and heavy defense equipment.

Designing for Welding Improv ap

of Heavy Duty Min

This close-up, left, of the upper section of the boom shows its type of construction. The lateral tubing members are welded to the toe of the chord angles. Diagonal diaphragm members are also steel tubing and are fitted into the inside corners of the chord members.



Revolving base plate, at right, is completely arc welded with the main bed plate, side stands, cross separators, and boom foot sockets forming one integral unit. This type of construction avoids entirely the use of bolts to hold parts together. All machining operations are performed with one setting of the frame in the machining fixtures.

Lightweight welded boom increases the live load capacity of this machine, left, by one-half cubic yard. Boom is sectional and designed for lengthening to suit conditions or dismantling for portability. Chord members are low-alloy, high-tensile steel angles and the lateral members are alloy steel tubing, a design which effects a 40 per cent saving in weight over conventional booms.

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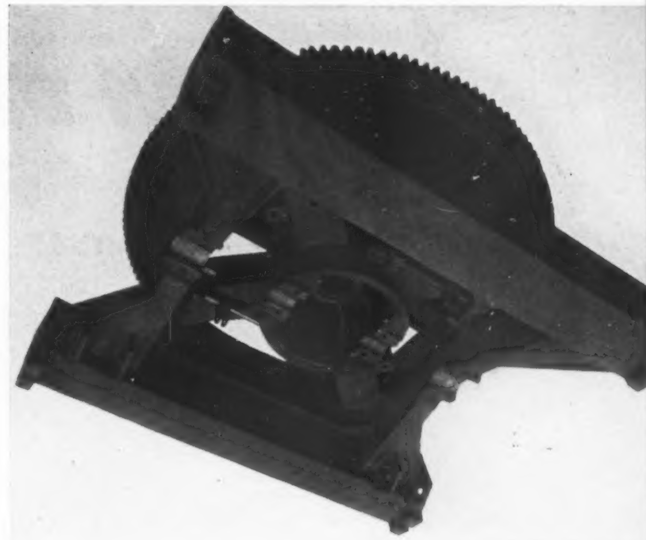
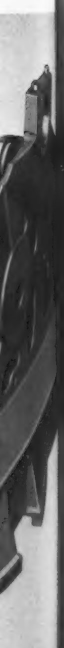
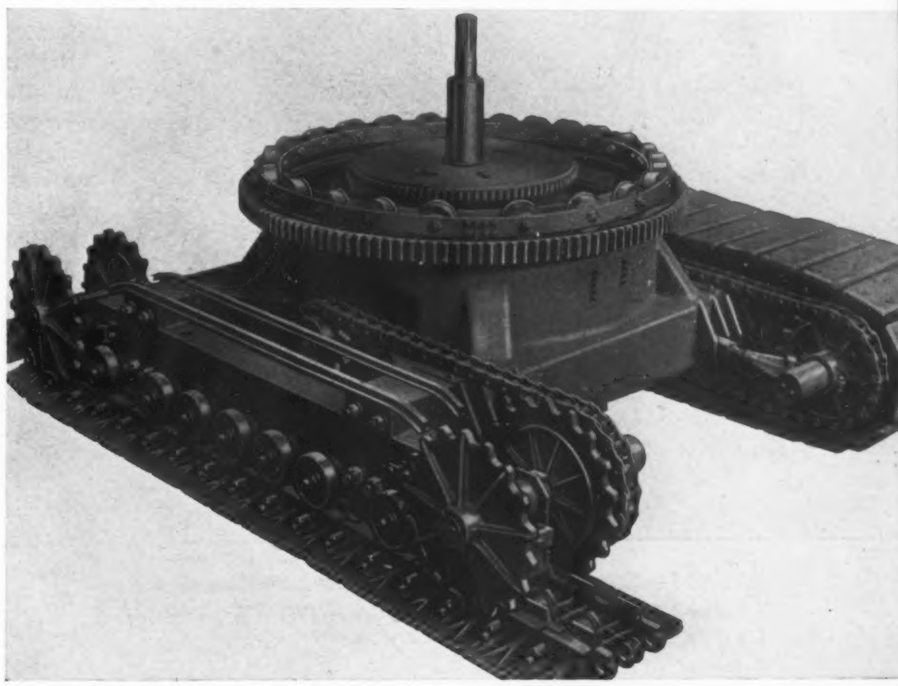
M ine

Bottom view, at right, of carbody shows the manner in which the loads are carried between the axles. Bearings are cast steel and securely welded in place before machining. The frame is composed of simple elements whereby the main vertical load over the roller track is carried on a barrel section which rests on box-type axles projecting from one crawler frame to the other. Center gudgeon is carried in a central barrel-shaped section which is held in place by H-beams welded to the inside cylinder and also connected to the large cylindrical section.

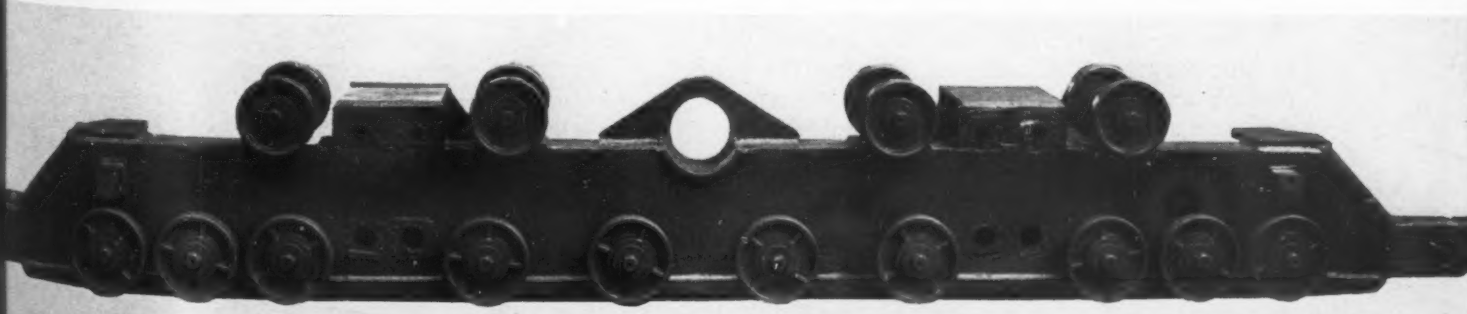




Lower structure of welded carbody, at right, with track, gear and roller track in place. Frame is unit welded with track frames bolted to the main frame. Design departs radically from previous practice on frames of this character which are cast steel. Details of construction and method of carrying load are shown in the illustration of the underside view of carbody.



Crawler frame, below, is designed for bolting to the main frame. Main section is box construction of channels turned toe-to-toe. Weight reduction is about 40 per cent over cast steel frames.



Specifying DESIGN DEPARTMENT MATERIEL

Part IV—Reproduction Equipment

By H. T. Pentecost

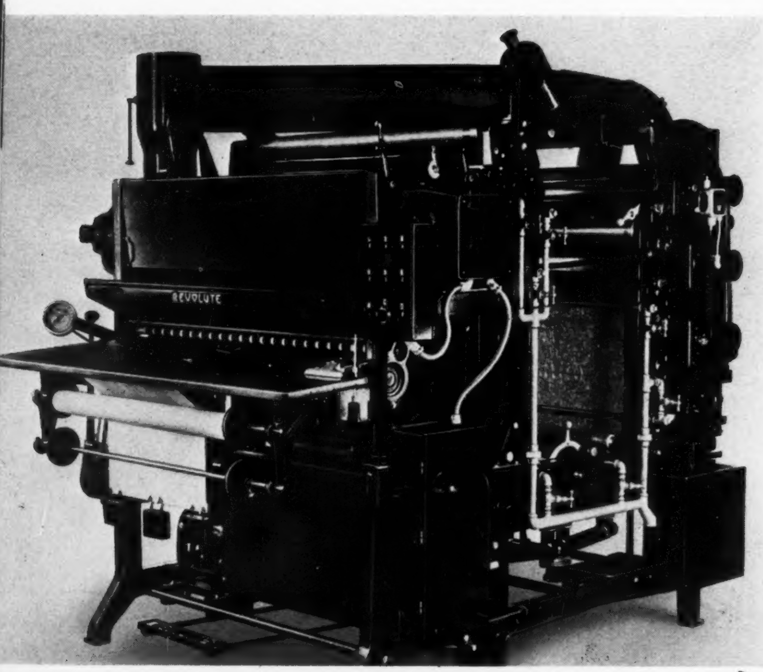


Fig. 1—Unit reproduction machine has attachments for white prints as well as van dykes. Drum for exposure rotates with the paper

PRINT-MAKING machines, to which this article is devoted, are the means by which the engineering department maintains contact with manufacturing. Equipment to expedite the production and handling of engineering department drawings formed the substance of the three previous articles on this subject. The reproduction or print is, however, the ultimate end toward which the making of drawings is directed. Bottlenecks in this department are no more to be condoned than in a production line. To meet today's unprecedented work schedules, reproduction equipment fortunately is available which for speed, quality and economy, establishes a new high.

One machinery manufacturing company still makes blueprints by exposing, in a glass frame, the superimposed drawing and paper to either sun or incandescent light, whichever is brighter. Developing and washing in improvised tanks and drying the resulting print by draping it tastefully over a steam radiator completes the process. It is a far call from such methods as these to the modern unit machine which prints, develops, fixes, washes and dries, producing flat, wrinkle-free reproductions at speeds up to and including 27 linear feet minute.

Two such machines are shown in *Figs.* 1 and 2. Slight differences, as in indicated maximum speed, exist among the several available machines of this type. However, reproduction speed depends to a large extent on the quality of the drawing or tracing—the opacity and sharpness of lines and transparency of the paper as well as the "speed" of the sensitized paper itself.

In one outstanding point, the design of the two machines illustrated differ. This is in the sliding or nonsliding contact of the papers during exposure and drying. Both systems have advantages in ironing out wrinkles and paper irregularities on one hand and eliminating static electricity and

possible smudging on the other.

Machine illustrated in *Fig.* 1 is equipped with an attachment for making contact, direct process prints. Also, for use in reproducing van dykes, a hypo bath is incorporated in addition to the blueprint potash bath. Either bath may be instantly selected by means of a convenient operating lever, as shown diagrammatically in *Fig.* 3.

Regardless of instructions and precautions pertaining to the use of prints in the shop, the design engineer cannot be sure that sooner or later someone will not scale dimensions from a print. In the case of assembly or subassembly drawings, there is a strong suspicion that this also happens occasionally even in the making of detail drawings.

That difficulties will arise out of such misuse of prints is, of course, inevitable. However, it would be unwise for the design engineer to take official cognizance of the fact that scaling of dimensions was being done. Recognizing the evil as being present in some cases, the fact that, in the design of modern printing machines, careful attention has been given the avoidance of stretching, wrinkling, or otherwise distorting the print, minimizes errors from this source.

Can also Reproduce Tracings

Attention should be given the broader scope of these machines in being able to make prints from which additional blueprints may be made. These prints are made on waterproof, transparent, sensitized tracing cloth. Not only is the retracing of pencil drawings thereby eliminated but the process also permits original drawings to be filed in fireproof vaults, and reproducible prints to be supplied to separate departments or branch plants.

Machines discussed to this point which incorporate in one unit light exposure, developing and drying find their most suitable application in large engineering departments where the volume of printing is high. Ideally, they should be used where it is possible to run an entire roll of paper, thus eliminating the necessity of frequent "rethreading."

Fig. 2—Printing machine is capable of exposure, developing and drying at speeds up to 27 feet per minute

In many smaller engineering departments such machines are not justified. Here, machines of the type illustrated in *Fig.* 4 provide a satisfactory solution to the printing problem. Exposure to light is their sole function. Separate facilities must be available for developing and drying. Prints can be exposed from a roll of paper as in the unit machine or they can be made singly. Whereas speed and efficiency of the overall printing job have been necessarily sacrificed to some extent, quality is unimpaired. Low cost, attractive appearance and the attention given the minimizing of operating noise

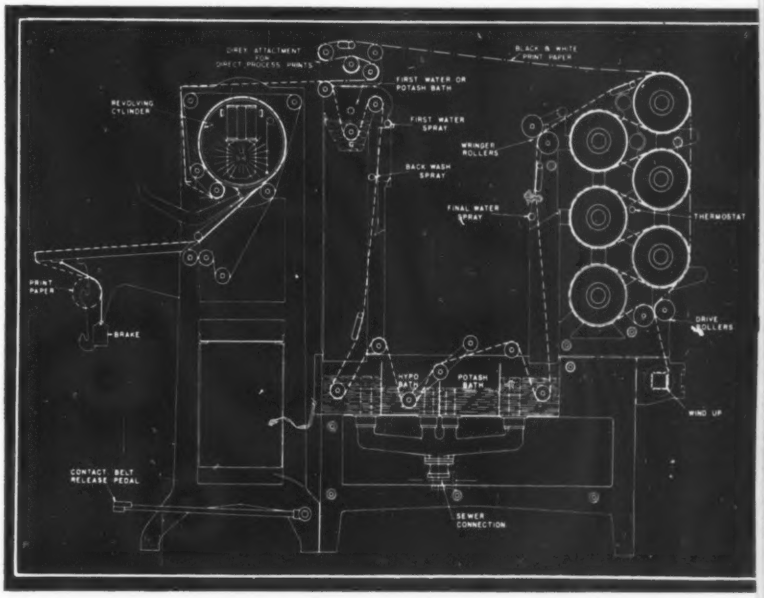


Fig. 3—Process diagram of machine illustrated in Fig. 1 shows method of changing from hypo to potash bath

are compensating features of these modern machines.

Finding their most extensive application in the category which may be termed special printing operations are machines which utilize a vacuum to hold tracing and sensitized paper in intimate contact. When prints are desired from unusually fine-lined tracings, or where tracings are aged, creased and brittle to the point where they are endangered by even the mild stress imposed by their passage through any of the previously discussed machines, these vacuum printers are invaluable. Exposure is accomplished by mounting the carbon-arc light sources in a carriage supported on rails. The light source is traversed across the vacuum frame, providing a progressive light exposure of the paper.

Whereas no additional equipment other than that already discussed is essential to the reproduction of so-called white prints (dark lines on a white

Fig. 4—Printing machine for exposure to light only. Careful attention has been given noise reduction, enabling machine to be installed directly in department

background), the increasing adoption of this process in many engineering organizations makes advisable a brief summary of its features. White prints are direct contact reproduction, made without use of negatives and requiring neither washing nor drying. Hence no shrinkage is experienced with the result that the sensitized paper is supplied in cut sheets in sizes corresponding to standard drawings in use in the department. Since the background is white, notes and alterations can be easily made without recourse to special writing media.

When, for purposes of record or for any other reason, permanent reproductions are desired, photocopying machines similar to that illustrated in *Fig.* 5 possess a valuable utility. In addition to full size reproduction, enlargements up to twice original size or reductions to half size are possible.

If greater reductions than half size are required, these also are possible by means of simple machine adjustments. Instead of placing the original on the copy board, it is mounted on an easel or wall board in such a manner that its distance from the lens may be materially increased.

A ground glass focusing screen is built into the machine so that it can be swung down into position for use. This glass shows the exact image of the copy to be photographed and hence can be easily checked for legibility.

Focusing dials are geared by means of a rack and pinion to the slidable camera carriage and copy board. Thus exact duplication of enlargement or reduction can be made by merely ascertaining that the dial readings, in each case, are identical.

Since this process is fundamentally photographic, the original drawing need not be on translucent paper or cloth. From an original, negative prints (white lines on black background) are made. From these, by the simple expedient of reprocessing, positive prints (black lines on a white background) are made.

Illustrated machine, the so-called duplex, produces copies on both sides of specially sensitized paper. For ordinary copying work, a commercial machine is available which, lacking the revolving magazine, copies on only one side. As shown in *Fig.* 5, a washing and drying attachment may be supplied with the photocopier, facilitating the making of reproductions.

Light Exclusion Unnecessary

In operation, the original drawing is placed on a copy board which is adjustable vertically to obtain the desired enlargement or reduction. Exposure, printing and fixing then take place within the machine. Prints are either removed for washing and drying or, if the available attachment is used, these operations are performed automatically by the unit. Thus no dark room is required.

MACHINE DESIGN acknowledges with appreciation the assistance of the following companies in the preparation of this article: Charles Bruning Co.; The Haloid Co., *Fig.* 5; Ozalid Products Co.; Paragon Revolute Corp., *Figs.* 1 and 3; C. F. Pease Co., *Fig.* 2; The Frederick Post Co.; Wickes Brothers, *Fig.* 4.

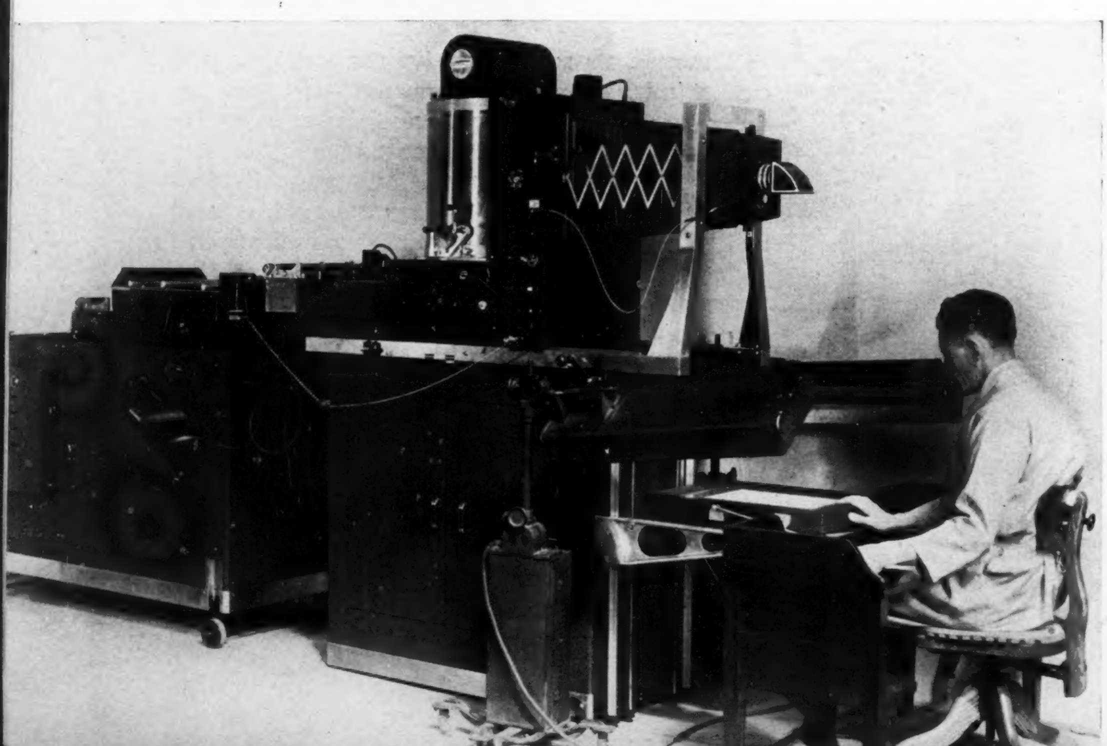
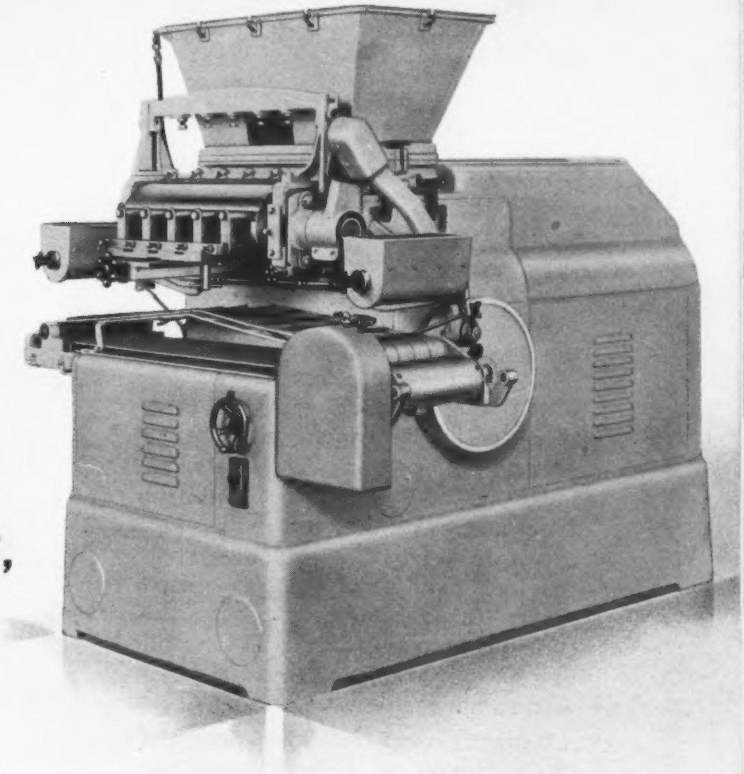


Fig. 5—Shown equipped with automatic washing and drying attachment, photocopying machine can make either enlargements or reductions

By J. C. Jones
American Bakers Machinery Co.



Obtaining Flexibility, Accessibility with Unit Assemblies

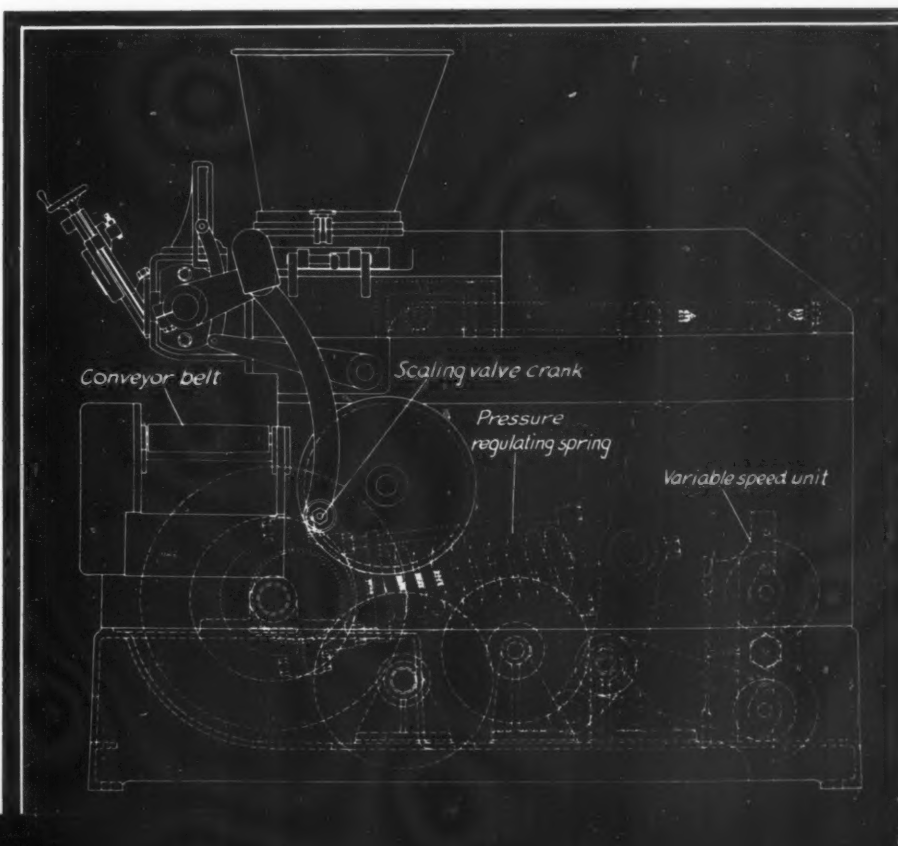
Fig. 1—Food machinery is designed to facilitate cleaning and to meet varying conditions

Fig. 2 — Below — Sectional view shows principles of operation

SIMPLE design, reliable operation, compact yet accessible parts, and attractive appearance are features that new designs should incorporate. An example of progress in that direction is the dough divider illustrated in *Fig.* 1. All parts of the machine are so disposed that the cleanliness required for food handling equipment may be maintained easily. To facilitate cleaning, the drive and food handling units are effectively separated. This feature has an additional advantage in that unit construction of assemblies allows changing of head, replacement of units or other modifications without altering the drive unit.

Purpose of this machine is to cut bulk doughs into pieces of equal weight for further processing in the forming into loaves. Because the material is practically of uniform density and simple handling methods are desirable, volumetric measurement is employed. To provide this measurement, a spring-regulated pressure piston, as shown in *Fig.* 2, places the material into scaling pockets. This is accomplished by drawing and separating pieces from the general mass into a compression chamber from which the scaling pockets are loaded.

In earlier dividers, the force required to move the dough through



the machine and to load the scaling pockets was secured through the use of a counter-poised weight. This did insure that uniform force would be applied at the compression chamber and on the dough during the loading of the pockets but, with the lighter doughs, it became excessive. The condition was corrected with the more recent design by using a spring in the operating connections to provide regulated pressure. A simple adjustment for piston travel is shown in drawing, *Fig.* 4, to provide the handling of the required quantity of dough for each stroke. This regulation eliminates the recompression that would be harmful by retarding the dough in its later development.

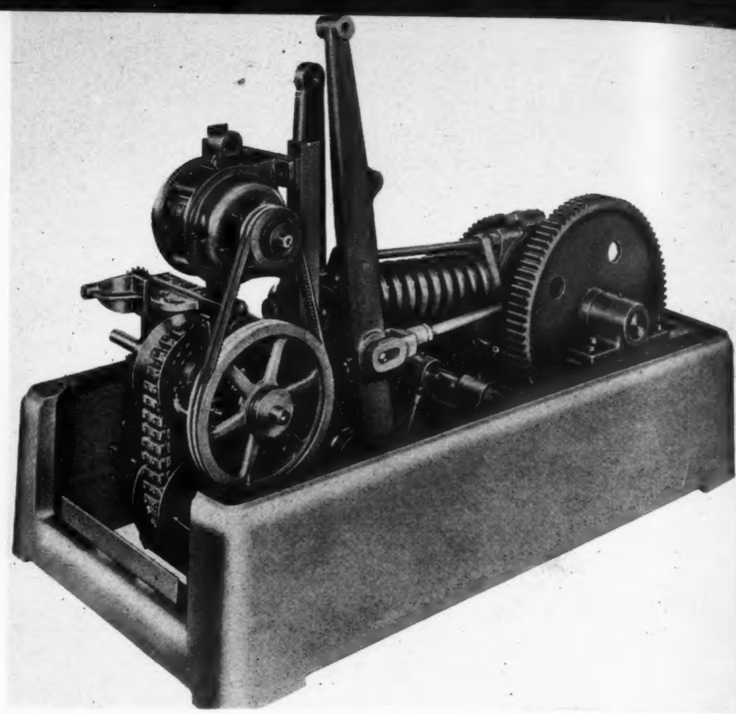
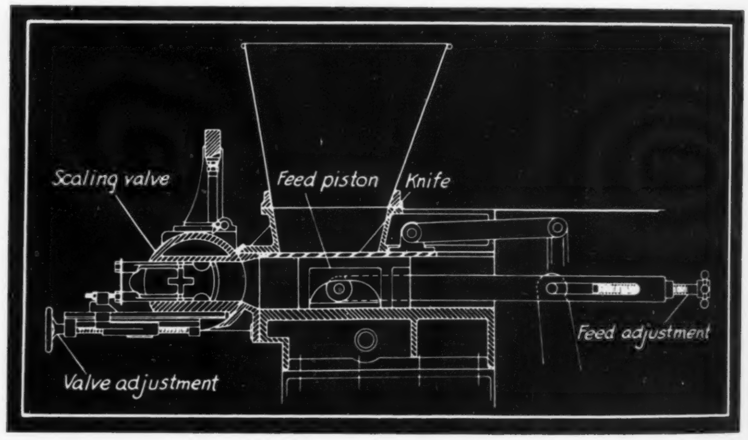


Fig. 3—Drive mechanism is mounted on heavy base casting, simplifying maintenance, lubrication, alignment

With the completion of loading, the scaling valve is turned from the horizontal to the vertical position. Dough pressures are maintained until the ports from the compression chamber have been closed by the cylindrical valve surface. The discharge mechanism then forces the plungers into the scaling pockets, discharging the pieces to the separating belt below the valve. This scaling valve is carried in oilless bearings to eliminate the possibility of lubricants being dropped on the dough pieces or the separating belt. Because pieces of different weight and size are handled, provisions are made for variable travel of the separating belt to insure the proper spacing and placing of the succeeding groups of loaves on the discharge belt.

Fig. 4—Below—Detail diagram of piston chamber and scaling valve. Divider lubricant is supplied all parts for moving or cutting dough



Drive includes a variable speed transmission with all operating parts mounted on a single, heavy base casting, *Fig.* 3, insuring rigidity and alignment. Also this simplifies assembly, maintenance and the installation of lubrication lines to distribution points. Possibility of lubricant contact with the food product being handled is thereby precluded. Standard commercial parts are used for the motor connection and for power transmission. Because these parts actually form a separate assembly, it is simple to arrange the machine for either constant or variable speed operation.

Side Plates Support Head

Side plates are mounted direct on the base casting. In addition to forming the frame, they are designed to support properly the head assembly and discharge conveyor. They are extended with a frame construction which permits complete enclosure, providing a modern design with pleasing appearance and lines that facilitate cleaning.

In the actual scaling operation, to insure the free flow of dough through the machine and for the protection of operating parts, special divider oil is regularly fed from a displacement pump to hopper, moving parts of compression chamber, scaling pockets and the valve. Delivery may be adjusted both with respect to quantity and pressure. Convenient means of regulating is provided at the points of discharge. Pump feed is controlled by an adjustable bypass relief valve to insure adequate delivery at required pressure to all points on the line. The individual controls permit the regulation of flow to the actual point of discharge. These controls, as all others, are immediately available to the operator of the machine.

With respect to accessibility for cleanliness, the design permits the swinging back of the receiving dough hopper mounted on the head of the machine. This affords access to the compression chamber to permit the removal of the blade or knife and the compression plunger. There is free access also to all parts coming in contact with the dough and the passages through which it travels.

Applying Theory of Elasticity in Practical Design

Part VI

Concentrated Loads

By R. E. Orton
Chief Engineer
Acme Steel Co., Chicago

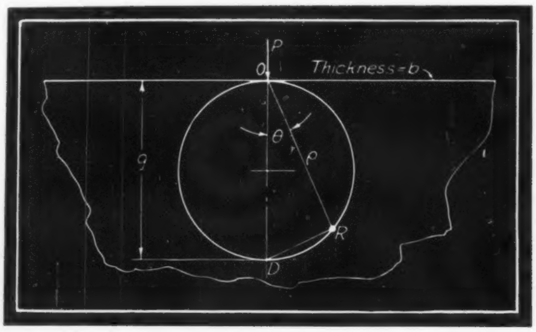


Fig. 48—Concentrated load applied normally to a semi-infinite plate. Stress along circle OD is constant

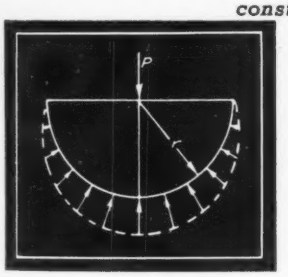
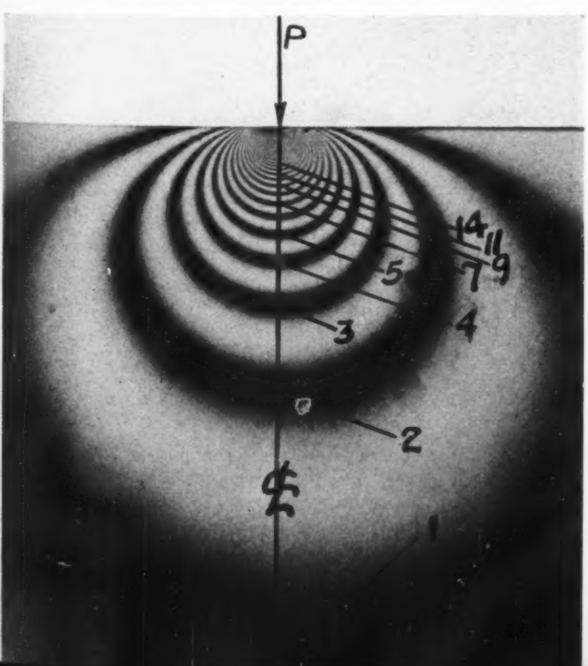


Fig. 49—Left—Stress on a semicircle must balance load P

Fig. 50—Below—Photoelastic picture of concentrated load on a large plate. Pattern closely approximates the semi-infinite plate. Enlarged 4.25 diameters



MANY loads are applied to the elements of a machine by contact between two parts. The contacting bodies may be in the form of rollers or other curved surfaces approximating a concentrated load or may consist of contact over a definite, relatively small area, producing a "partially distributed" load. Such loadings give rise to stress disturbances of a more or less local character. An understanding of the nature of these stresses is a material aid in the interpretation of photoelastic pictures and in the design of critically stressed parts. While in many cases the actual problem will not meet theoretical requirements, it usually will be possible to make an approximation sufficiently close for commercial work.

Where the load is applied to other than a circular shape the solution is usually difficult and has not been offered in all cases. Typical is the column load illustrated in the photoelastic picture of *Fig.* 52. The problem may be greatly simplified, in the case of loading on a plane, by consideration of the theoretical shape of a plate extending indefinitely in all directions from the loaded plane. Such a hypothetical shape is known as a "semi-infinite" plate. It may be relatively thin, giving plane stress; or it may be "thick" or long, giving plane strain. As demonstrated before, the equations to be developed apply to either case.

Depends on Nature of Stress

The applicability of the solution of the semi-infinite plate rests on the local nature of the stress. Accordingly, to insure this local nature in a piece of finite dimension, the load must be distributed over an area that is small relative to the extent of the loaded plane, such as may be considered to be the case in the column of *Fig.* 52 at points close to the load. In such a practical problem the theoretical solution will give a good approximation or, at least, will serve as an excellent guide to photoelastic solution.

This article discusses the solution for a concen-

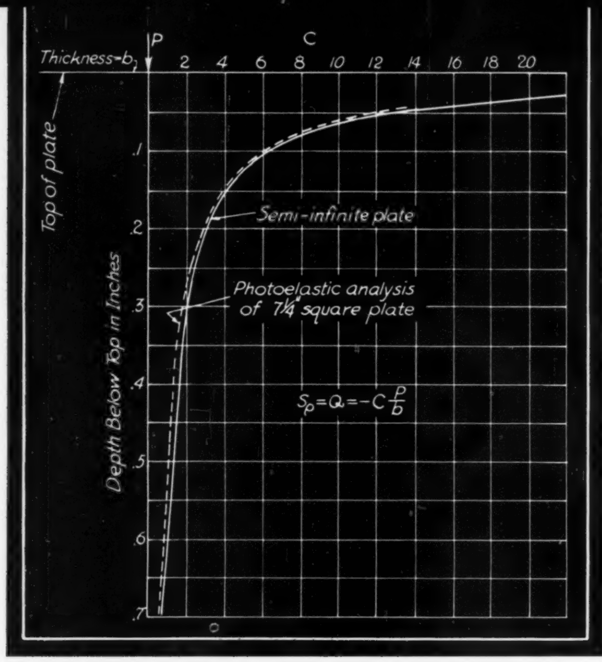


Fig. 51—Stress along center line of semi-infinite plate from a concentrated load

trated load and a partially distributed uniform load applied to the boundary of a semi-infinite plate. Succeeding articles will discuss a concentrated load applied at any inclination to the end of a wedge, and concentrated loads applied to a disk or roller.

As was discussed in the column analysis of Part III (M. D., April), a concentrated load is only a theoretical consideration because a load applied over an area of zero width would necessitate a material of infinite rigidity. Therefore, the stresses in the *immediate* vicinity of the load will depend on the deformation occurring between the contacting bodies. The discussion of this deformation and of the load distribution resulting from this deformation is reserved for a later article. At distances away from the vicinity of the load, by St. Venant's principle (Part III), the load may be assumed as concentrated. This is the second stress region referred to in connection with the column analysis.

In *Fig.* 48 is illustrated a concentrated load applied perpendicularly to the boundary of a semi-infinite plate. Angle θ is measured from the direction of the load with positive direction as shown. The stress function for this case is

$$\phi(\rho, \theta) = A\rho\theta \sin\theta$$

Substitution in Equation 66 (Part IV, M. D., May) shows it satisfied and therefore the stresses from this function are compatible. The stress system is obtained from Equations 63, 64 and 65

$$S_\rho = \frac{1}{\rho}\frac{\partial\phi}{\partial\rho} + \frac{1}{\rho^2}\frac{\partial^2\phi}{\partial\theta^2} = \frac{2A\cos\theta}{\rho}$$

$$S_\theta = \frac{\partial^2\phi}{\partial\rho^2} = 0$$

$$v_{\rho\theta} = -\frac{\partial}{\partial\rho}\left(\frac{1}{\rho}\frac{\partial\phi}{\partial\theta}\right) = 0$$

The first boundary condition is that when $\theta = \pm\pi/2$, $S_\theta = v_{\rho\theta} = 0$ which is obviously met because these stresses are zero for all values of θ. The second condition is that when $\rho = 0$, S_ρ must balance P. At this value of ρ, S_ρ becomes infinite. Since the stress is distributed over an infinitely small area, the equation takes on the indeterminate form $\infty \times 0$. It is necessary, therefore, to solve this load balance in some other way. This may be done by considering a semicircular portion of radius r, centered about the origin, shown set out as a "free body" in *Fig.* 49. S_ρ is given by

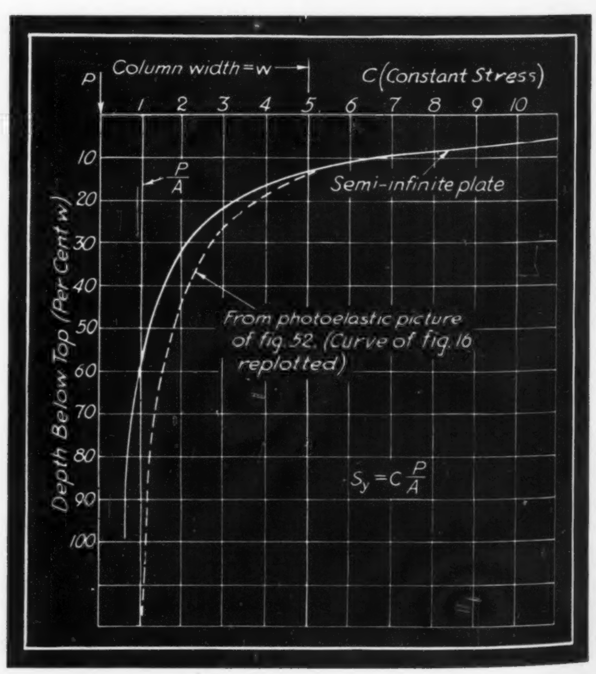
$$S_\rho = \frac{2A}{r}\cos\theta, \qquad \text{for} \qquad \rho = r$$

The vertical components of this stress, summed up over the boundary, must balance P

$$\int_{-\frac{1}{2}\pi}^{\frac{1}{2}\pi} (S_\rho b r d\theta)\cos\theta = -P$$

Fig. 52 — Right — Pressure ring fringes are invariably present in contact loading, even on relatively small members as shown in column loading. Enlarged 3.9 diameters

Fig. 53—Below—Column stress close to load is approximated by solution for semi-infinite plate



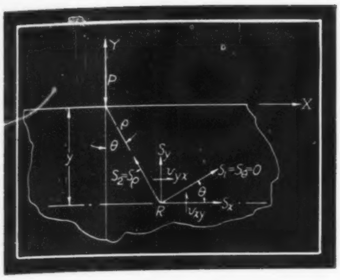


Fig. 54 — Left — Stress directions in rectangular co-ordinates for concentrated load

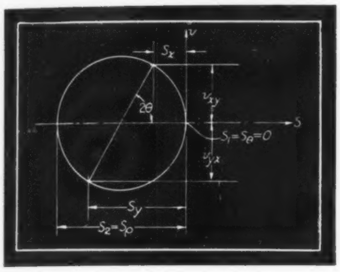


Fig. 55 — Left — Mohr's circle diagram for stress distribution shown in Fig. 54

Substituting for S_ρ, integrating between the limits, and solving for A

$$A = -\frac{P}{\pi b}$$

Substituting for A in the stress equations, gives finally for the stress system*

$$S_\rho = \frac{-2P\cos\theta}{\pi b \rho} \qquad (105)$$

$$S_\theta = v_{\rho\theta} = 0 \qquad (106)$$

The principal directions are radial and tangential and the tangential stress is zero. The system is one of simple radial compression. As ρ approaches zero the stress becomes infinite. In order to meet the requirements of a finite value of the elastic limit, it may be considered that a semicircular notch of sufficient radius to give a stress below the elastic limit is cut out at the origin, and this notch loaded so as to balance the stress along its boundary.

In *Fig.* 48 a circle of diameter g has been drawn with its center on the extension of P, and tangent to the boundary at the origin O. Since the angle formed by the radius vector to any point R on the circle, and the line RD, subtends a diameter it is a right angle. Therefore, the value of the radius vector is given by $\rho = g\cos\theta$. Substituting in Equation 105 gives

$$S_\rho = \frac{-2P}{\pi b g} \qquad (107)$$

The stress, then, is constant at all points on the circle. Since S_θ is zero, the stress difference Q is equal to S_ρ. The photoelastic picture will consist of a series of ring fringes centered on the extension OD of P, and tangent to the boundary at O. The fringe value, n, is given by $n = Qb/H$, Equation 45 (Part III, M. D., April) where H is the calibration value of the model material. Substituting and solving for g gives for the diameter of the "pressure rings"

$$g = \frac{2P}{\pi H n} \qquad (108)$$

Because the diameters of the rings are given by integral values of n they will be spaced in a harmonical progression, becoming closer and closer as the load is approached. All the rings pass through O, which gives the anomaly of a point having various stress values. Mathematically this is explained by the indeterminate form that Equation 105 takes on at this point, $S_\rho = 0/0$. Practically it may be explained on the ground that the load must actually be distributed over some small distance, causing a departure here from Equations 105 and 107.

While the theoretical problem cannot be exactly reproduced in a photoelastic analysis, it may be closely approximated by the use of a relatively large plate. *Fig.* 50 is a photoelastic picture of a 7¼-inch square plate centrally loaded through a ⅜-inch diameter steel pin. Plate is fully supported along its lower boundary to eliminate bending stress. $P = 96.8$ pounds, $b = .272$-inch, $H = 87$. Though the first order fringe is distorted somewhat due to the proximity of the side and bottom boundaries, the rings come closer to a circular shape as the load is approached. The two parallel scratches near the bottom are to determine the enlargement.

In *Fig.* 51, the stress along the vertical "center line" has been plotted from Equation 107. The corresponding stress, taken from the photoelastic picture of *Fig.* 50, is plotted in the dotted curve. The

Fig. 56 — Right — Stress distribution for an inclined load is the same as for a normal. θ is measured from the direction of P

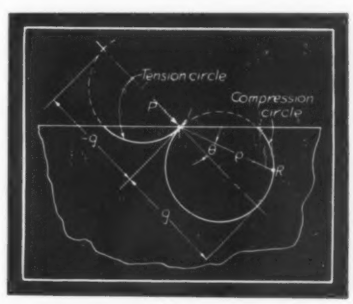
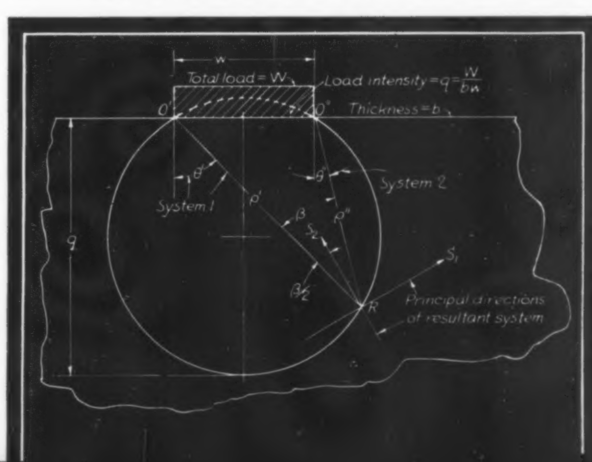


Fig. 57 — Below—Stress distribution from a partially distributed uniform load is built up on two sets of co-ordinates noted as Systems 1 and 2



* The original solution of this problem is credited to J. Boussinesq and Flamant, Paris, 1892. It was suggested by the photoelastic work of Carus Wilson, 1891.

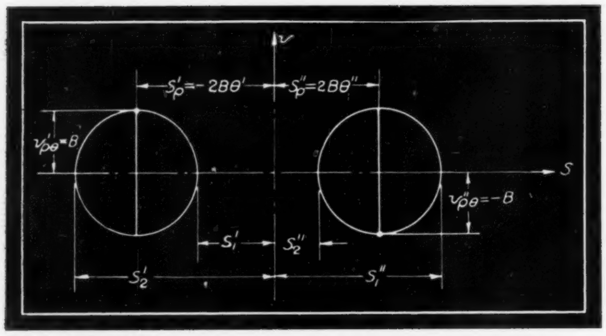
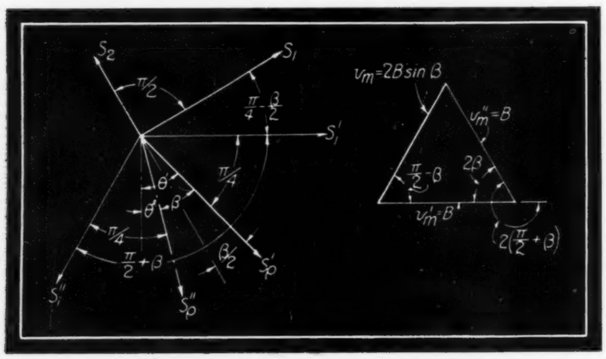


Fig. 58—Mohr's circle for the two stress systems of Fig. 57 for a partially distributed uniform load

Fig. 59—Below—Stress directions and composition of the pure shear elements of Figs. 57 and 58



photoelastic work checks the theoretical within 3 to 4 per cent at points close to the load. The first order fringe is within 7 per cent of its theoretically indicated position, although the plate width is only ten times the fringe diameter.

Characteristic pressure-ring fringes are present in contact loading even in many cases where the extent of the "plate" is small. This is strikingly brought out in *Fig.* 52. The rings are here distorted to an elliptical shape due to the proximity of the sides of the column. As the load is approached they become smaller and approximate more closely a circular shape. The notch-like appearance immediately under the load is due to plastic deformation.

Calculations Agree Closely

In *Fig.* 15, the stress along the center line of this column is plotted, as taken from the photoelastic picture in *Fig.* 52. This curve is reproduced as the dotted curve of *Fig.* 53. The solid curve is the stress as computed from Equation 107. It may be noted how closely it approaches the photoelastic curve near the top of the column, confirming the assumption that the center line stress in the "second region" may be closely approximated by this equation.

Frequently the load stress will be desired in rectangular co-ordinates. In *Fig.* 55 Mohr's circle diagram is shown for a point *R*, Fig. 54, located *y* below the boundary. The *X* and *Y* co-ordinate directions are taken as usual, *y* therefore being always negative. From *Fig.* 55, the stresses are

$$S_x = \frac{2Px^2y}{\pi b\rho^4} \qquad (109)$$

$$S_y = \frac{2Py^3}{\pi b\rho^4} \qquad (110)$$

$$v_{xy} = \frac{2Pxy^2}{\pi b\rho^4} \qquad (111)$$

where $\rho = (x^2 + y^2)^{1/2}$. Both S_x and S_y are always compressive, v_{xy} is positive on the right of the center line, negative on the left. An interesting check on the development of these equations is made by substituting in the basic equilibrium equations 1 and 2 (Part I, M. D., Feb.). Also, examining the boundary conditions, all three stresses approach zero as the distance from the origin increases without limit (x and y appear to the third power in the numerator while ρ is to the fourth). All three stresses are zero at the finite boundary $y = 0$.

These equations are particularly valuable for the approximate analysis of finite bodies. In the discussion in the April issue of the column of *Fig.* 52, the principal directions were assumed vertical and horizontal at the center line. This assumption was based on Equation 111 whch shows $v_{xy} = 0$ there

Fig. 60 — Right — Mohr's circle for the resultant of the stress systems of Figs. 57, 58 and 59

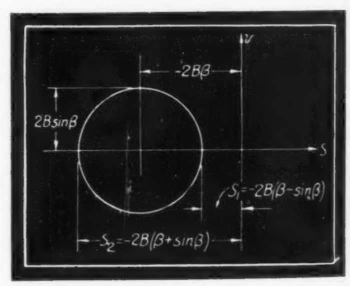
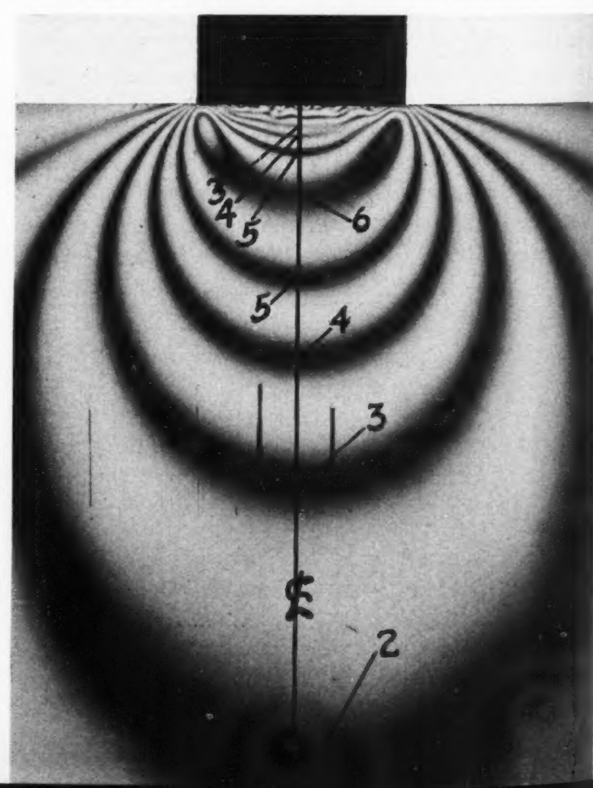


Fig. 61 — Below — Photoelastic picture of partial uniform load on a large plate. Pattern approximates the semi-infinite plate. Enlarged 2.2 diameters



and on the uniform distribution prevailing in a column width away from the load. It is justified by the isoclinic picture of *Fig.* 20. Similarly, from Equation 109 and the column condition, it was concluded that S_x was zero at the center line. This is justified by the closeness of the two curves of *Fig.* 53.

From Equations 109, 110 and 111, the middle portion of the curves in *Fig.* 16, showing the relative distribution across the column section, were constructed. From the partial derivative with respect to x, some idea of the slope that occurs at the center line is obtained. The remainder of the curves are constructed from the boundary conditions at the sides of the column, $S_x = v_{xy} = 0$ and from the equilibrium of a section of the column which shows that the average value of S_y must equal P/A.

If the concentrated load P is inclined instead of vertical, the development and stress equations in polor co-ordinates are the same, provided θ is again measured from the load direction, as shown in *Fig.* 56. The stress will be tension when θ is greater than 90 degrees. Stress rings will appear above the origin, opposite hand to the others, with simple tension instead of compression.

The stress distribution from a partially distributed uniform load applied normally to the boundary of a semi-infinite plate is obtained by the superposition of two systems developed on different coordinate positions. *Fig.* 57 illustrates the load of intensity q applied over the width w. The origins of the two stress systems are located at either end of the load, with positive directions as indicated on the figure.

System 1 is developed from the stress function

$$\phi' = -B(\rho')^2\theta'$$

Substituting in Equation 66 shows that the stresses from this function are compatible. Substituting ϕ' in Equations 63, 64 and 65 gives for the stresses

$$S_\rho' = S_\theta' = -2B\theta'$$

$$v'_{\rho\theta} = B$$

System 2 is developed from the function

$$\phi'' = +B(\rho'')^2\,\theta''$$

which, from its similarity to ϕ', is obviously also compatible. The stresses, similarly, are

$$S_\rho'' = S_\theta'' = 2B\theta''$$

$$v''_{\rho\theta} = -B$$

It now remains to combine these two systems to obtain the resultant stresses. This will be done by the use of Mohr's circle diagram, as described in the first part of the photoelastic series, MACHINE DESIGN, March 1940. The diagrams for these two stress systems are shown in *Fig.* 58. The "hydrostatic heads" are equal to the radial stress in this

Fig. 62 — Above — Superimposed light and dark field photos of loading for Fig. 61. Light field pictures give half step fringes. Enlarged 2.2 diameters

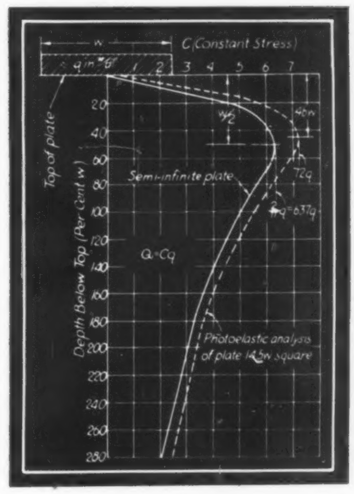


Fig. 63 — Right — Stress difference Q along center line of semi-infinite plate from partial uniform load

case since $S_\rho = S_\theta$. Adding and noting that $\theta' - \theta'' = \beta$, the angle included between the radius vectors to the point R gives for the head of the combined system

$$S_\rho' + S_\rho'' = -2B(\theta' - \theta'') = -2B\beta$$

The pure shear stresses remaining are now to be combined. Working from the radial directions and remembering that for Mohr's circle counterclockwise rotation of the shear stress is positive, the principal stress S_1' is $\pi/4$ in a counterclockwise direction from S_ρ', see *Fig.* 59. The shear stress of System 2 being negative, S_1'' is $\pi/4$ in a clockwise direction from S_ρ''. In turn, S_ρ'' is β clockwise from S_ρ'. Therefore the angle between S_1' and S_1'' is $\pi/2 + \beta$, as shown in *Fig.* 59. Also *Fig.* 59 shows the vectoral composition of the shear stresses, from which the direction of the resultant principal stresses, S_1 and S_2, are obtained. It should be noted that S_2 bisects the angle β between the two radius vectors. From the vectorial composition of *Fig.* 59 the resultant shear stress equals $v_m = 2B \sin \beta$.

Hydrostatic head and shear stress are now combined to form the Mohr's circle of *Fig.* 60. From this the principal stresses of the resultant system

(*Continued on Page* 122)

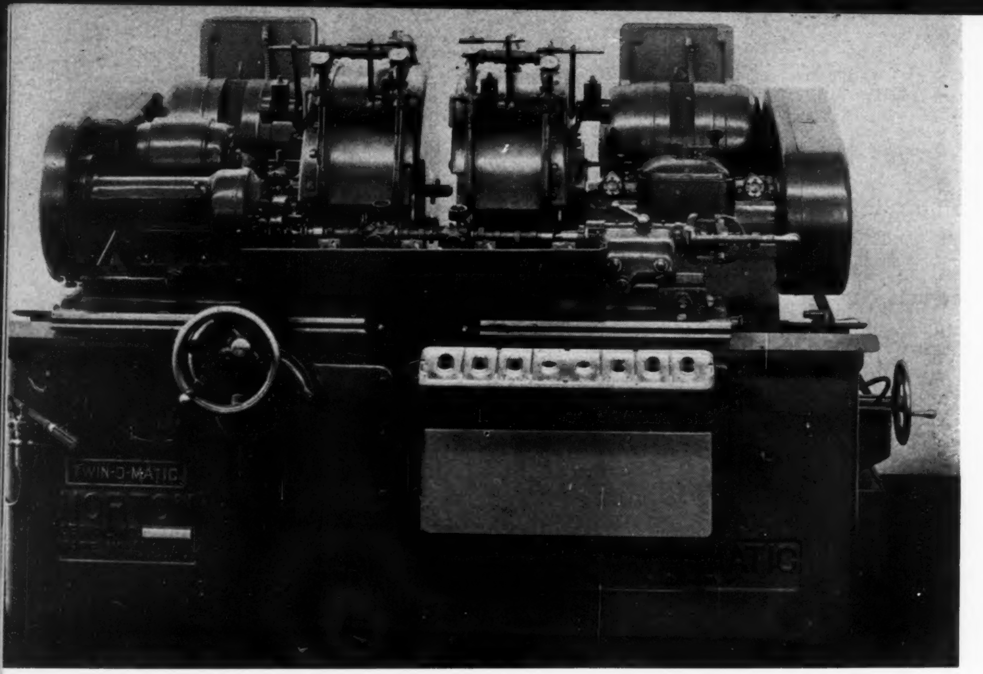


Fig. 1—Short-time ratings have advantages for special applications. In this grinder for cams, the work cycle is short

Selecting Speci

Part VII—Short Time Rated Motors

By G. R. Anderson

Chief Engineer, Electrical Division
Fairbanks, Morse & Co.

IN ANY consideration of the application of short-time rated motors it is essential that a clear understanding be established as to what is meant by the rating of a motor. Motors must have a rating to give them a standard dimension just the same as weight, area, or length is used for quantitative measurement. It is, therefore, universal practice to rate them on the basis of horsepower, speed, temperature rise and time of operation. It is also an axiom that when time is reduced, horsepower or temperature may be increased. The question naturally arises in the applications of motors to machines as to the factors to consider in order to effect economies from these criteria. Thus, in *Fig.* 1, short-time rated motors are applied advantageously to a special machine for grinding cams wherein the load cycles are short.

Uses Torque, Temperature and Time for Rating

The horsepower factor is an ideal and acceptable measurement for rating motors of the constant speed class. If, however, a motor is designed and built principally to serve an application involving torque and acceleration, then horsepower becomes an inadequate factor to express its rating. Under these conditions the basic factors become torque, temperature and time.

Other items that inherently influence short-time rating of motors are the type of protective enclosure, the method of cooling and relative size.

CONSTANT-SPEED GENERAL-PURPOSE CLASS: Contrary to the usual belief there are few advantages in short-time rated motors for constant-speed, general-purpose applications. A machine that operates intermittently or infrequently, having load characteristics where starting torque is not important, might conceivably be satisfactorily operated by a short-time rated motor. Such a motor would be of special design and probably built in a frame size or two smaller than a continuous-duty motor. If the smaller shaft size, bearings and frame are of sufficient strength for the particular load, a special motor of this class can be built at some saving over the continuous rated motor. The amount depends primarily on the time rating. The chances are, however, that this same intermittent load could have been satisfactorily handled by the next size

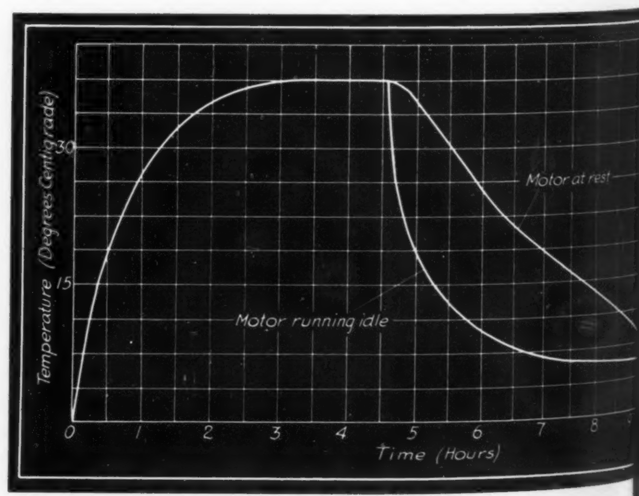


Fig. 2—Typical heating and cooling curves of a motor show difference in cooling with motor running idle and at rest

Motors

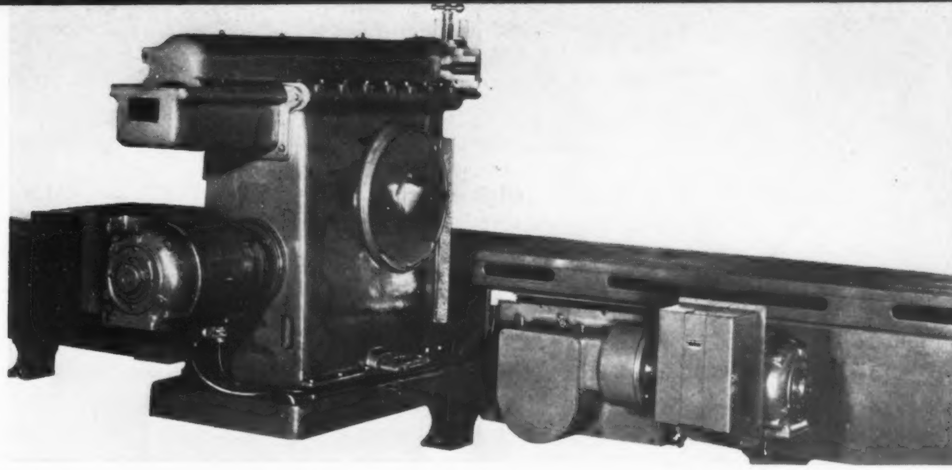


Fig. 3—Work and tool movement on special shaper advantageously utilizes short-time rated motors

smaller horsepower standard motor with a more appreciable saving in cost and with all the advantages of using a standard motor. Relative size of motor frames are shown for comparison in *Fig.* 4. Practically all constant speed, continuous duty motors are capable of carrying substantial overloads for short periods of time.

Reviewing the question of short-time rated constant-speed general-purpose motors from engineering design and manufacturer's viewpoint, the economy to be gained from a special design, short-time rated motor depends primarily on the time and temperature rating that can serve the application. A two-hour rated motor is practically no different from a continuous rated motor. A one-hour rated motor or a thirty-minute rated motor can usually be built in a frame size smaller than a continuous duty motor. A five, ten or fifteen-minute rated motor can be built considerably smaller and, by taking advantage of higher permissible temperature rise, the size can be further reduced. Any further reduction in time rating brings in the factors of starting and what the motor does in its idle time.

Requires One-Quarter Idle Time

How can short-time rating be determined? One simple way, if the load is constant, is to chart the time load curve of the machine and determine the maximum time for the heaviest load. Then, if the intervening periods of idle running or stopping are at least 25 per cent duration, this maximum time under load is a fair indication of the minimum short-time rating applicable.

Short-time ratings are generally accompanied by higher temperature ratings. This basis is sound in principle for it is generally accepted that life of insulation is a function of operating time as well as temperature. Therefore, short-time rated motors may usually be specified with higher temperature rise, such as, 50 degrees Cent. instead of 40 degrees Cent. as for continuous rated open type motors.

Another factor of importance is the effect of off-load cooling. In general, motors subjected to intermittent loading will operate cooler if run idle during off load, than will be the case if the motor is stopped. This can readily be verified by referring to typical motor heating and cooling curves illustrated by *Fig.* 2. Cooling is much more effective with the motor running idle than when stopped. Economies obtained from short time rated motors may be further improved by operating this way. Exception should be made in the case of totally enclosed motors without external ventilation.

TORQUE APPLICATIONS: Short-time rated motors serve their most useful and economical purpose for those applications where speed-torque characteristics are modified to suit the needs of a particular application. Loads involving starting and acceleration problems, plugging or reversing, may often be admirably suited for a short-time rated motor.

An ideal example of a short-time rated motor with characteristics modified to develop high starting and accelerating torques are squirrel-cage, elevator and hoist motors. Open motors for this service are usually rated thirty-minute duty, 50 degrees Cent. temperature rise. Full advantage is taken in the design of these motors by the nature of the duty cycle that requires intermittent high starting torque but automatically provides periods of light loads or rest.

Squirrel-Cage Motors for Torque Use

Since motors for special torque applications offer the best opportunity to take advantage of short-time rating, a brief analysis of factors to be considered in selecting squirrel-cage motors for reversing service will serve to illustrate the important point that *motors can be designed with a short-time rating to operate efficiently and economically on a continuous repeating duty cycle.* In other words, the motor characteristics may be such that it requires a short-time rating based upon horsepower and temperature but it will do its job continuously because of the nature of the duty cycle. An alternate method of rating a special motor of

this kind is to give it a higher temperature rating on a continuous duty basis.

In general, motors with special electrical characteristics are required when the application involves reversing service. Two methods can be employed to rapidly stop the rotation of a polyphase induction motor prior to reversing its direction of rotation. One involves the use of magnetic brakes and the other involves the reversal of power connections to the motor terminals. When the latter arrangement is used, the term "plugging" applies.

General purpose motors for reversing service: Squirrel-cage motors with general purpose characteristics are not suitable, as a rule, for reversing service except when the number of reversals is relatively infrequent. Further, they should not be used for reversing service where the starting load or load WR^2 is relatively high.

Where a load requires a few reversals per minute but at the same time has long periods of operation in one direction of rotation under load, then high-torque, low-starting-current, general-purpose motors provide the characteristics most suitable, provided that the periods of the reversing duty are not of too long a duration. In other words, reversing three to five times per minute in cycles not exceeding five or possibly ten minutes duration might be considered satisfactory for a high-torque, low-starting-current motor. This general statement should, however, be modified by the size and physical enclosure involved.

Special motors for continuous reversing service: Where applications involving continuous reversing service are encountered such as tapping machines, reversing planers, and shaper beds, *Fig.* 3, special motors are usually applied. These motors may be designed for continuous reversing; however, the maximum number of reversals per minute with normal temperature rise will depend to a large extent upon the load conditions.

To determine the possible maximum reversals per minute that can be obtained satisfactorily necessitates providing the motor manufacturer with complete load information as well as the data on voltage, phase and frequency of the power supply. Load requirements of the complete duty cycle should be furnished together with the WR^2 and speed of all rotating parts subject to reversal or the weight and size of each part either in rotating or linear motion. This information will enable the manufacturer to determine more accurately the size and rating of the motor and the maximum permissible reversals per minute that such a motor can withstand.

Minimum Speed for Reversing Best

In general, where motors are to be applied to reversing service the speed of the motor should be kept to a minimum. Special reversing motors are designed with characteristics such that the average torque developed during reversed direction of rotation and accelerating in its connected direction of rotation is obtained with an average minimum value of loss in the motor. The running full-load speed of motors designed for reversing service is considerably lower than the full-load speed of a general-purpose continuous duty motor.

Where the prime work of the motor is taken up in the energy necessary to accelerate and decelerate, it is likely that the motor will be designed with a

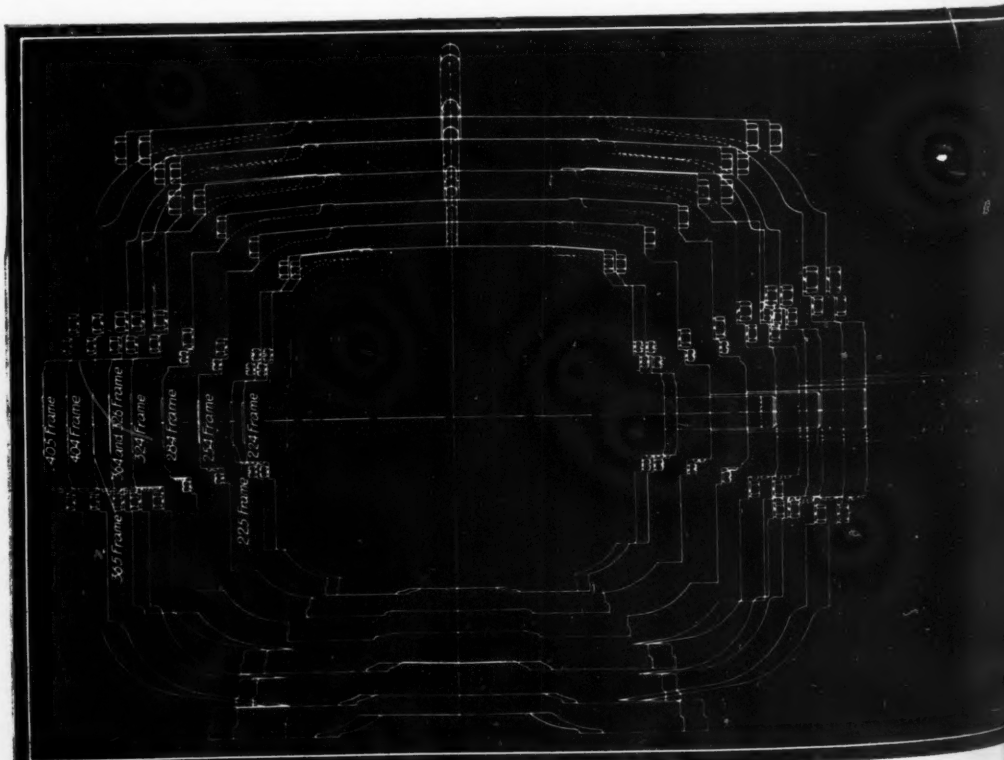


Fig. 4—Outline drawing of NEMA motor frame sizes shows graphically the relations between the different frames. It is useful in laying out space requirements for a motor. Reduction scale is 6:1

Fig. 5—Short-time rated motor designed for exceptionally severe hoisting and stalled torque application has a continuous repeating cycle. With continuous, externally supplied ventilation, size is reduced and the application of a squirrel-cage motor made practical

high slip and even though it may operate continuously, its horsepower rating may necessarily be specified for short-time duty. Use of class *B* insulation is also a factor in keeping down the size of the motor for reversing service, particularly when the WR^2 of the rotor may absorb a large proportion of this energy. It may even be found expedient, on applications of this nature, to design the motor in a relatively small diameter to accomplish economy in size and power required.

Effect of Enclosure Depends on Time

Rating as affected by enclosures: The degree of short-time rating applicable for different conditions is also influenced by the type of frame enclosure. Most of this discussion has been based upon the use of open motors. Splash-proof, enclosed fan-cooled, or totally-enclosed frames affect, to a considerable degree, the time rating that can be given to a special motor designed for modified torque applications. There is relatively little difference between the splash-proof motor and the open motor. The enclosed fan-cooled frame makes more difference and in the case of a totally-enclosed motor, the degree becomes considerably more marked. As an example of the effect of enclosure, where a high torque motor for elevator or hoist service is rated thirty-minute duty, 50 degrees Cent. rise when built in the open frame, the same motor would be rated fifteen-minute duty, 55 degrees Cent. rise when built in an enclosed frame.

On the other hand, if the degree of short-time rating is extremely small such as five or ten-minute duty, the type of enclosure begins to make little difference. In motors that operate for about a minute at a time, such as those employed for opening and closing valves or similar application, the degree of enclosure makes still less difference in relative size of the motor.

Size is also a factor in determining the economy of short-time ratings. In general, the larger the rating, the greater will be the relative economy particularly if the time duty is short.

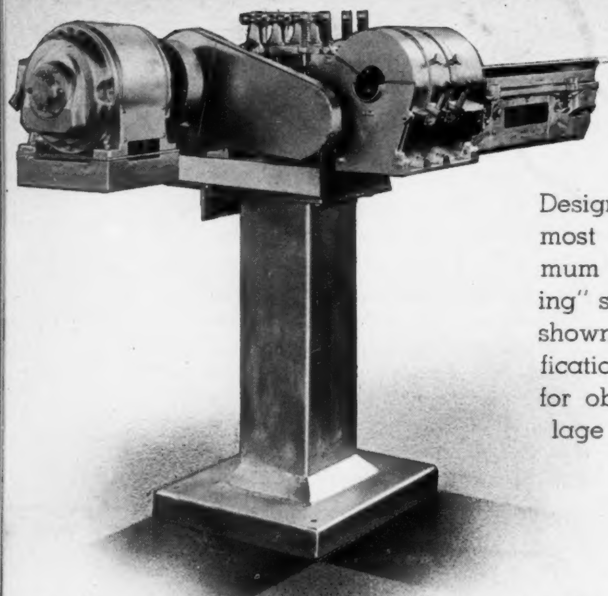
There are many typical application examples where short-time motor ratings are used to advantage. It will be noted that, in nearly all, the factor of torque is paramount in the requirements of the load. Typical examples of short-time rating applications in the order of thirty-minute and one-hour duty are elevators, hoists, coal loading conveyors, car pullers, and laundry extractors.

Typical examples of short-time rating applications in the order of 5 to 15-minute duty are valve operating motors, baling presses, anchor winches, hoists, door opening motors, and torque motors.

There are occasional applications that impose such a severe duty on the motor and are of a continuous repeating nature that not only must the size of the motor be held to a minimum but external means for continuous ventilation be provided. *Fig.* 5 illustrates a motor designed for an exceptionally difficult application that combined hoisting and stalled torque in a continuous repeating cycle. The addition of a continuous flow of air from an external source made the application of a short-time rated squirrel-cage motor practical.

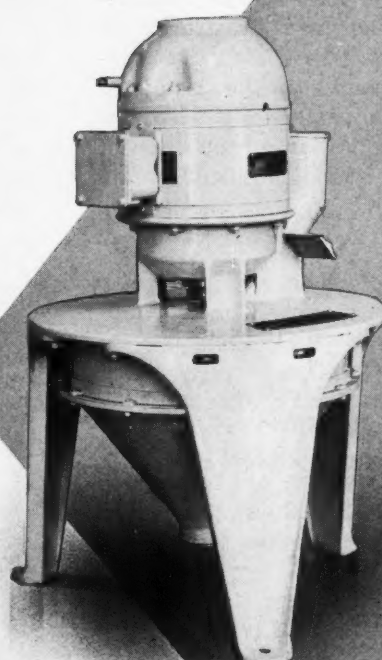
Summarizing briefly, a few of the important points in the application of short-time rated motors are:

1. There are few advantages in short-time rated motors for constant-speed, general-purpose applications
2. There are many advantages in the use of short-time rated motors for squirrel-cage motor applications requiring modified speed-torque characteristics, particularly where high starting torques are required
3. For short time applications, off-load cooling is more effective with motor running than when at rest
4. Short-time rated motors can, in many cases, be designed and applied to loads having a continuous repeating duty cycle
5. Best results are obtained on squirrel-cage motors applied to reversing service, which may be a continuous repeating duty cycle, when the speed is kept to a minimum.



Designed for short-range, liaison flights, foremost performance requirement is low minimum flying speed, approaching the "hovering" speed of autogiros. Flaps and wing slots shown in the Bellanca (Right) meet this specification. A high-wing monoplane, visibility for observation service is unimpeded. Fuselage and control surfaces are fabric covered

Perfect alignment of corn ears being fed to the first of the rotary cutting heads is achieved by use of a double set of rubber feed rolls insuring against damage to kernels. Drive is by means of a single V-belt; automatic centrifugal slip clutch is used to provide smooth starting and running. Food Machinery Corp. corn cutter (Above) has automatic, centralized lubricating system requiring a minimum of operator attention

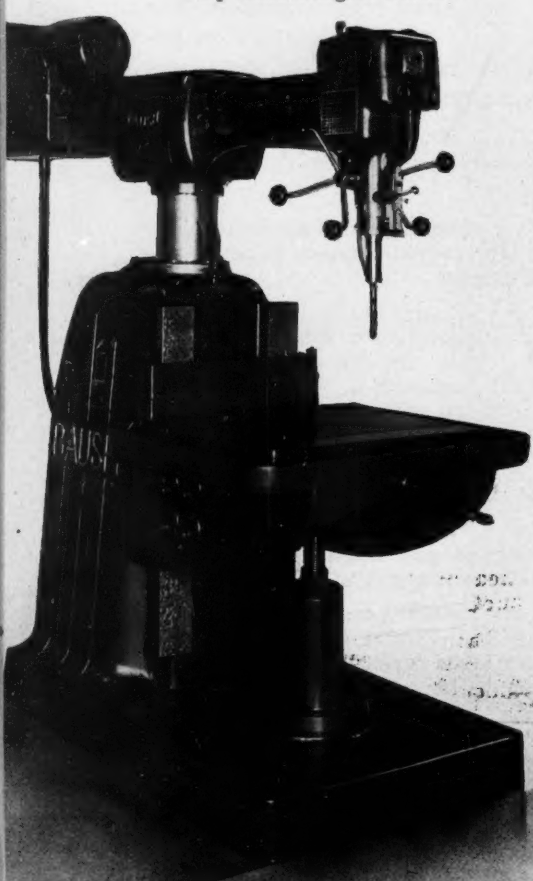


Design tur
IN NEW HI

(For new machi ge 136)

THIS MONTH'S COVER mounte ball bearing centers, i the cu in the National Broach Co. gea machine tool. Thus th torque in smaller machines wh drives gear, is eliminated. C built fo axes settings and is pro e bar c so that any setting may licated. of developing the ellipti machin gears up to 2 meter

Rotated at 3500 revolutions per minute by a direct-connected, special-purpose electric motor, the rotor of the Entoleter infestation destroyer (Right) destroys insects, larva or insect eggs which may be present in flour. Rotor is composed of two plates between which the flour or meal is fed. Impactors in the form of cylinders connecting the plates act as baffles which destroy insects when they are thrown against these cylinders at high velocity by centrifugal force



Automatic electric counting switch permits drilling of holes to a predetermined depth and tapping of holes flush to shoulders in the Baush drilling and tapping machine (Left). Advance of tap is positively controlled by lead screw. Conversion of head from one operation to another is facilitated by clutch. Disk brake type, ¾-horsepower motor provides drive through variable transmission

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V HINES

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COVER mounted in free
ers, is the cutting tool
roach Co. gear shaving
mus the torque, present
es wh drives the work
d. C built for crossed
is prov e bar adjustment
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ellipt machine handles
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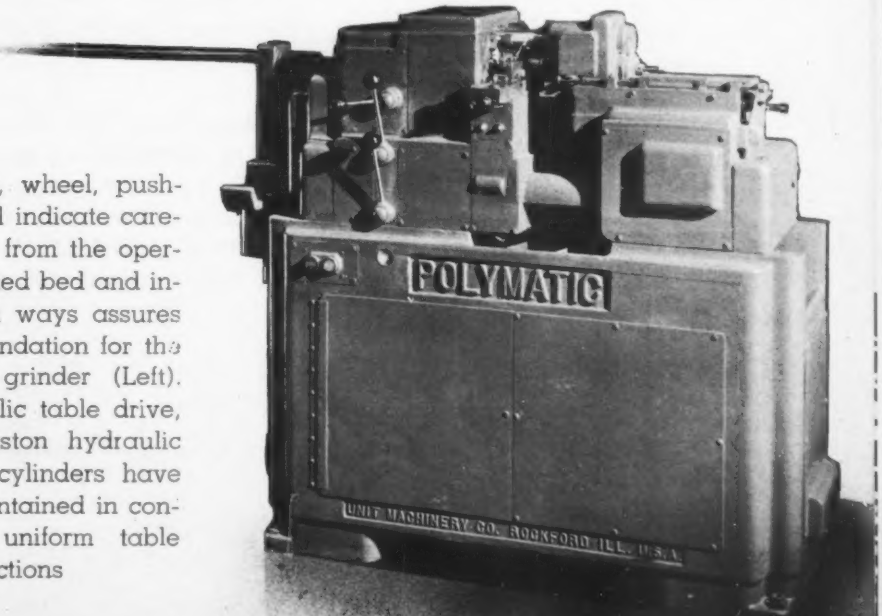
One-piece delivery drum and automatic, coin-operated trapdoor are the only moving parts in Slector bottle beverage dispenser (Right). Flat type cooling plates of the dry refrigeration process insure trouble-free operation. High and low-load refrigerating requirements are met by automatic expansion valve and ¼-horsepower, overload-protected motor driving compressor unit

Incorporating an L-shaped heat distributor providing 40 per cent more heat distributing surface than conventional drum models of comparable size, efficiency and capacity are correspondingly increased. Porcelain enamel finish on both inside and outside of the Norge space heater (Left) facilitate cleaning and prevent corrosion. Air is drawn down to the flame through a radial orifice which imparts to it a turbulent, whirling motion

Heat treated, alloy steel helical gears, cross shaved and lapped, constitute the drive of the Polymatic screw machine (Below). Base is heavily ribbed cast iron on which is mounted, on an adjustable platform, the drive motor. Location of the front bearing close to the spindle nose provides exceptionally rigid work support. Provision is made for close adjustment of double-row, precision roller bearings

Location of control lever, wheel, push-button and instrument dial indicate careful attention given design from the operator standpoint. Lengthened bed and increased distance between ways assures an exceptionally rigid foundation for the table of Diamond face grinder (Left). Noteworthy is the hydraulic table drive, powered by a rotary piston hydraulic pump. Double-opposed cylinders have piston rods which are maintained in constant tension, assuring uniform table speed in both directions



MACHINE

Editorial

DESIGN

Designers' Greatest Responsibility Is Still To Come

IN THE year since the inauguration of the defense program the country's designers have met a challenge never before imposed. Armament equipment almost undreamed of prior to the fall of France has moved swiftly from the creative and development stages to the production lines. And that the effect of this almost superhuman effort in designing and producing against time is making itself felt is only too evident from the fact that the theaters of war are, at the moment, turning eastward.

This diversion, as in the case of less major diversions of their military efforts in the past, undoubtedly is planned as another step in the attainment of the aggressor nations' ultimate goal. It leaves unabated the "challenge to design" with which our engineers are faced. There is strong reason to feel that the second year of the defense program may well call for even greater effort than the first, in the production of effective military equipment. Design—and particularly armament design, under current conditions—is never static. It is up to us to develop planes, tanks and guns that, based on actual operation of existing equipment on battle fronts, is so far ahead of existing designs that the eventual issue can never be in doubt.

Achievements of designers too often have gone unsung. That applies particularly to the period immediately preceding the commencement of hostilities. In the recent past, however, hardly a day passes in which some national authority does not credit engineers with playing a major role in the country's welfare. Times without number the war has been referred to as a conflict between machines rather than men, and this assertion has been emphatically confirmed. It is not the number of men a nation can put in the field that spells success, but the superiority of the military equipment used.

Again the burden falls on design—assuming that in the future production is less likely to be hampered by government regulations, fifth column activities and strikes for petty grievances. That America's designers will continue to rise to the occasion and reach even greater heights of accomplishment is the confident belief of all who have taken time out to analyze the progress of this unprecedented "battle of machines."

Professional Viewpoints

MACHINE DESIGN welcomes comments from readers on subjects of interest to designers. Payment will be made for letters and comments published

"... is excellent analysis"

To the Editor:

Mr. Talbourdet's work on 4-bar linkages is an excellent example of basic analyses, and makes a fine contribution to the general subject of linkages. May there be many more of the same general type and caliber!

I have long believed that there are two major phases to the problem of machine design: First, the functional design; second, the production design. Functional design consists of the development, selection, and arrangement of various mechanical motions to accomplish a given purpose. This part of design is largely "inspirational." Production design starts with the critical examination of the functional design to develop the details so that they can be made economically. It also involves analysis of the motions so as to exploit their possibilities to the greatest extent. This part is largely analytical. These should be handled as two distinct tasks, even though the same person does them both.

As speeds increase, and other conditions become more critical, more precise analyses must be made of every mechanical element and motion. To do this with the minimum expenditure of time, it is necessary to develop general analyses of different mechanical motions so that the particular case can be solved by substituting the particular values and computing.

—EARLE BUCKINGHAM
Massachusetts Institute of Technology

". . . . selection is ideal"

To the Editor:

The inauguration of "Engineering Data Sheets" adds considerably to an already excellent magazine, and the selection of the "Four-Bar Linkage" is ideal to open a series of this nature.

Will these Data Sheets be prepared in book form at a later date? At any rate, I would like to get a reprint of the "Mathematical Solution of Four-Bar Linkages" for my files, if at all possible.

I would also like to get a reprint of the article, "Designing Snap Ring Fastenings."

—F. J. FURMAN, *Assistant Engineer*
International Business Machines Corp.

A limited supply of data sheets is available and copies will gladly be supplied to readers upon request.—ED.

"... vocational guidance is needed"

To the Editor:

The question raised by H. A. Bolz in your May issue is most timely. Vocational guidance properly organized and applied, not only by our educational institutions but also by individual engineers, would greatly increase the effectiveness of our industrial system.

Altogether too many of our young men find themselves starting upon a career for which they are not suited. The discouragement and loss of precious years of time which accompany such misfits are a distinct detriment to the progress and defense of our country.

More personal counsel by engineers with young men is needed. They should emphasize the necessary qualifications for an engineering career and help students avoid retracing their steps.

Many of our college students should be attending vocational schools because their natural abilities and talents lie in the direction of the manual arts, in which they would be more successful and consequently more happy.

—C. E. SCHIRMER, *Chief Engineer*
Robbins & Myers, Inc.

ASSETS *to a* BOOKCASE

Design of Machine Elements

By Virgil Moring Faires, Professor of Mechanical Engineering, Agricultural and Mechanical College of Texas; published by The Macmillan Co., New York; 490 pages, 6 by 9 inches, clothbound; available through MACHINE DESIGN *for $4 postpaid.*

This second edition contains numerous additions of new material and improvements throughout the book. Noteworthy are a new chapter on variable stresses and stress concentrations, and additional data on the properties of materials. Also included in the new sections are computations of stresses in welded connections, use of hydrodynamic theory in the design of thick-film journal bearing, dynamic load for the design of all types of gears.

Prepared principally as a text, it is useful also for reference work and review purposes. Covered as chapters are engineering properties of materials, stress analysis, tolerances and allowances, screw fastenings, shafts, gearing, bearings, springs, brakes and cams. As a supplement, a booklet containing problems and useful data is available for a nominal price of $1.40.

□ □ □

Matrix and Tensor Algebra

By Clarence E. Rose, Engineer, Navy Department; published by Chemical Publishing Co., New York; 143 pages, 5½ by 8¼ inches, clothbound; available through MACHINE DESIGN *for $4 postpaid.*

This book has been written so that anyone possessing the rudiments of differential calculus can readily interpret it. Because so many valuable technical discussions now use this method of analysis, it is necessary to know the elements of matrix and tensor algebra to keep abreast of developments in physics and engineering, both in their broad and specific fields. The author undertakes to enable the engineer to master the elements of this subject so he can read the books and papers applicable to his work without additional instruction. For the benefit of those with advanced knowledge, the earlier chapters may be omitted. For others, the repetition is not objectionable but serves instead to add emphasis.

Leading up to the subject are treatments connecting it with classical mathematics, vector analysis, determinants and solution of simultaneous equations. Matrix algebra discussion develops functions of matrices and their differentiation, integration, and mechanics. Tensor algebra chapters include addition, subtraction, multiplication, covariant, contravariant, mixed tensors, etc. Development of Christoffel's symbols in 2 and 3-space and in the 4-dimensional space-time continuation is presented in a clear interpretation.

□ □ □

Plastics in Industry

Published by the Chemical Publishing Co., Inc., Brooklyn, N. Y.; 241 pages, 5½ by 8½ inches, clothbound; available through MACHINE DESIGN *for $5.00 postpaid.*

Although the authors of the volume remain annonymous, the book discusses the various groups and types of plastic materials in an interesting manner. Chemical machinery and plant for the production of materials are covered to acquaint the reader with the characteristics obtainable in molding compounds. Molding and fabrication techniques are treated and special formulas for various applications and their particular specifications are included. Chapters are devoted to particular industries such as electrical, aircraft, automobile, textile, building, fancy goods, furniture, and packaging. Synthetic glues and synthetic rubbers are also given chapters.

□ □ □

Who's Who in Engineering

Published by Lewis Historical Publishing Co., New York; 2107 pages, 9 by 6 inches, clothbound; available through MACHINE DESIGN *for $10 postpaid.*

A fifth edition, this volume contains the professional records of over 15,000 engineers, an increase of 25 per cent over the last edition which appeared in 1937. As far as practicable, the editors have included all engineers of outstanding and acknowledged professional eminence, engineers with more than ten years practice and responsible charge of important engineering work, and outstanding teachers of engineering subjects.

Every listing is a corrected record or new biographical information, both types of which were supplied and approved by each individual listed. Included in the book, in addition to the alphabetical listing of outstanding engineers, are a geographical cross-index, listing of engineering organizations with officers and their total membership, and professional and honor fraternities listed similarly.

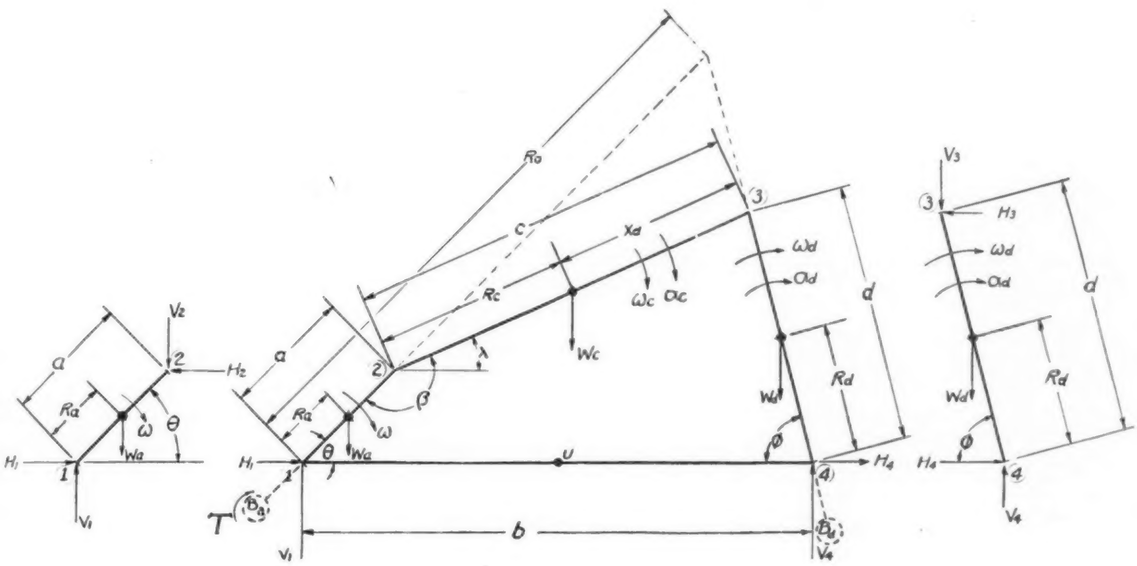
Mathematical Solution of 4-Bar Linkages

Part IV—Balancing of Linkages

By Guy J. Talbourdet & Paul R. Shepler

Research Division

United Shoe Machinery Corporation



$$R_o = \frac{b \csc \theta}{\cot \theta + \cot \phi} \qquad \lambda = \theta + \beta - 180^\circ$$

$$\omega_c = \frac{\omega\, a\, (\cot \theta + \cot \phi)}{a(\cot \theta + \cot \phi) - b \csc \theta}$$

$$\alpha_c = \frac{\omega^2 a\, R_o}{(a - R_o)^2} \left(-\cot \theta + \frac{\csc^2 \theta + \csc^2 \phi \dfrac{\omega_d}{\omega}}{\cot \theta + \cot \phi} \right)$$

Expressions for ϕ, $\omega_d = \dfrac{d\phi}{dt}$, $\alpha_d = \dfrac{d^2\phi}{dt^2}$ and β are given in the articles on 4-bar linkages in the May and June, 1941, issues of Machine Design.

$$V_R = V_1 + V_4 = -M_a R_a \omega^2 \sin \theta + M_d R_d \,(-\omega_d^2 \sin \phi + \alpha_d \cos \phi) - M_c \omega^2 a \sin \theta - M_c R_c \,(\omega_c^2 \sin \lambda + \alpha_c \cos \lambda) + W_a + W_c + W_d$$

$$V_1 = -M_a R_a \omega^2 \sin \theta - M_c \omega^2 a \sin \theta - M_c R_c (\omega_c^2 \sin \lambda + \alpha_c \cos \lambda) + \frac{I_{d_4} \alpha_d c \sin \lambda + (M_c R_c \omega^2 a \sin \beta + I_{c_2} \alpha_c) d \sin \phi}{c\, d \sin (\phi + \lambda)} + W_a + W_c \frac{X_d}{c}$$

$$V_2 = -M_c\, \omega^2 a \sin\theta - M_c R_c (\omega_c^2 \sin\lambda + \alpha_c \cos\lambda) + \frac{I_{d_4} \alpha_d c \sin\lambda + (M_c R_c \omega^2 a \sin\beta + I_{c_2} \alpha_c) d \sin\phi}{cd \sin(\phi+\lambda)} + W_c \frac{X_d}{c}$$

$$V_3 = -\frac{I_{d_4} \alpha_d c \sin\lambda + (M_c R_c \omega^2 a \sin\beta + I_{c_2} \alpha_c) d \sin\phi}{c\, d \sin(\phi+\lambda)} + W_c \frac{R_c}{c}$$

$$V_4 = M_d R_d (-\omega_d^2 \sin\phi + \alpha_d \cos\phi) - \frac{I_{d_4} \alpha_d c \sin\lambda + (M_c R_c \omega^2 a \sin\beta + I_{c_2} \alpha_c) d \sin\phi}{c\, d \sin(\phi+\lambda)} + W_c \frac{R_c}{c} + W_d$$

Note: Terms containing W may be omitted in calculations for low weight, high speed linkages.

$$H_R = H_1 + H_4 = -M_a R_a \omega^2 \cos\theta + M_d R_d (\omega_d^2 \cos\phi + \alpha_d \sin\phi) - M_c \omega^2 a \cos\theta + M_c R_c (-\omega_c^2 \cos\lambda + \alpha_c \sin\lambda)$$

$$H_1 = -M_a R_a \omega^2 \cos\theta - M_c \omega^2 a \cos\theta + M_c R_c (-\omega_c^2 \cos\lambda + \alpha_c \sin\lambda) + \frac{I_{d_4} \alpha_d c \cos\lambda - (M_c R_c \omega^2 a \sin\beta + I_{c_2} \alpha_c) d \cos\phi}{c\, d \sin(\phi+\lambda)}$$

$$H_2 = -M_c \omega^2 a \cos\theta + M_c R_c (-\omega_c^2 \cos\lambda + \alpha_c \sin\lambda) + \frac{I_{d_4} \alpha_d c \cos\lambda - (M_c R_c \omega^2 a \sin\beta + I_{c_2} \alpha_c) d \cos\phi}{c\, d \sin(\phi+\lambda)}$$

$$H_3 = + \frac{-I_{d_4} \alpha_c c \cos\lambda + (M_c R_c \omega^2 a \sin\beta + I_{c_2} \alpha_c) d \cos\phi}{c\, d \sin(\phi+\lambda)}$$

$$H_4 = M_d R_d (\omega_d^2 \cos\phi + \alpha_d \sin\phi) + \frac{-I_{d_4} \alpha_c c \cos\lambda + (M_c R_c \omega^2 a \sin\beta + I_{c_2} \alpha_c) d \cos\phi}{c\, d \sin(\phi+\lambda)}$$

$$T\ (Torque) = -V_2 a \cos\theta + H_2 a \sin\theta$$

$$M_u (Vibrating\ moment) = V_1 \frac{b}{2} - V_4 \frac{b}{2}$$

Procedure for Balancing Linkages

1. Balance driving and driven cranks around the center of rotation of each

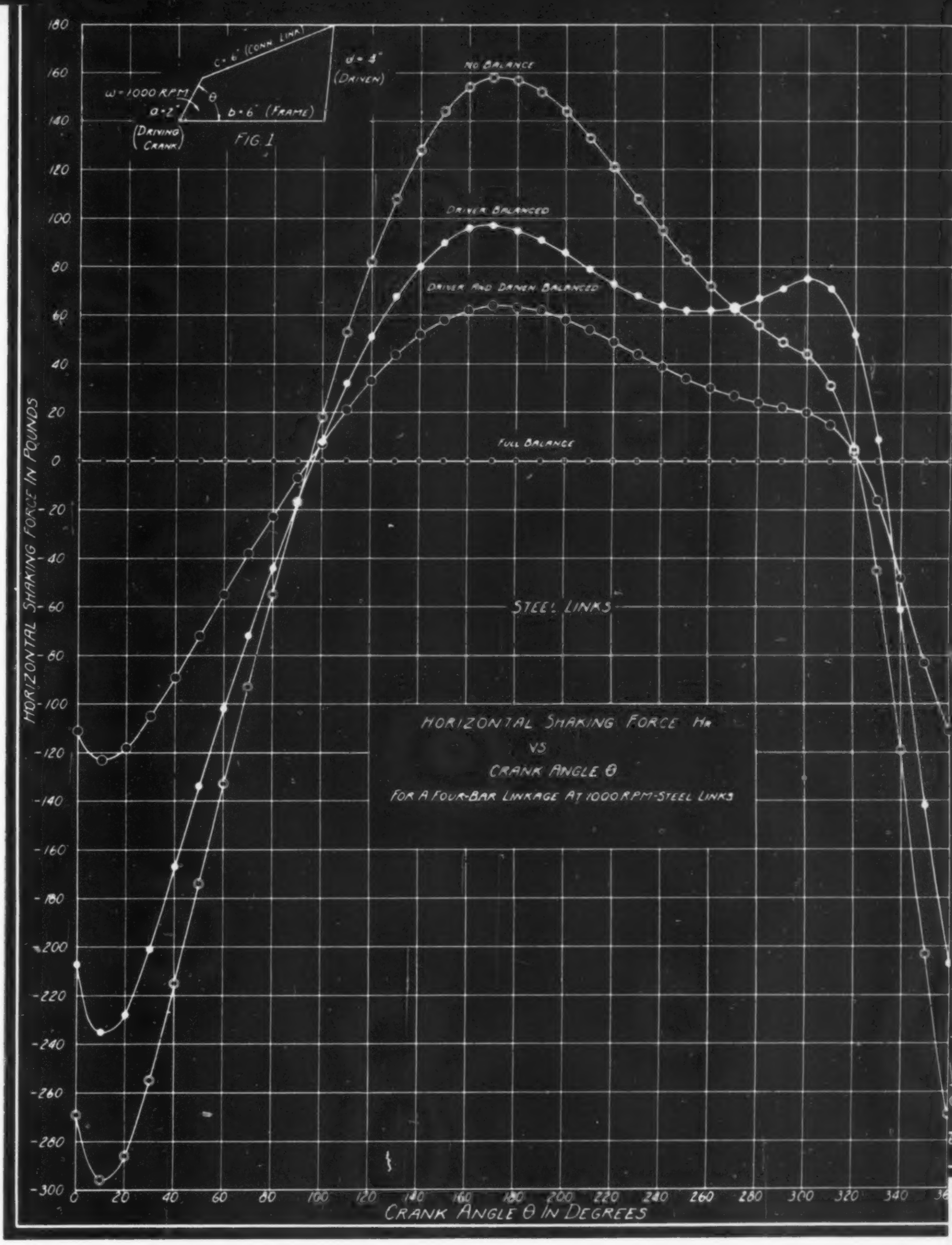
2. To balance effect of connecting link, add additional weights to each crank so that these weights will have the following first moments:

   $$M_{a_1} = \frac{a X_d W_c}{c}\ (weight\ B_a) \qquad M_{d_4} = \frac{d R_c W_c}{c}\ (weight\ B_d)$$

   Polar inertia of balance weights should be maintained as low as possible by keeping their centers of gravity close to the center of rotation. Centers of gravity of balance weights should lie as close as possible to center lines of cranks to eliminate dynamic unbalance

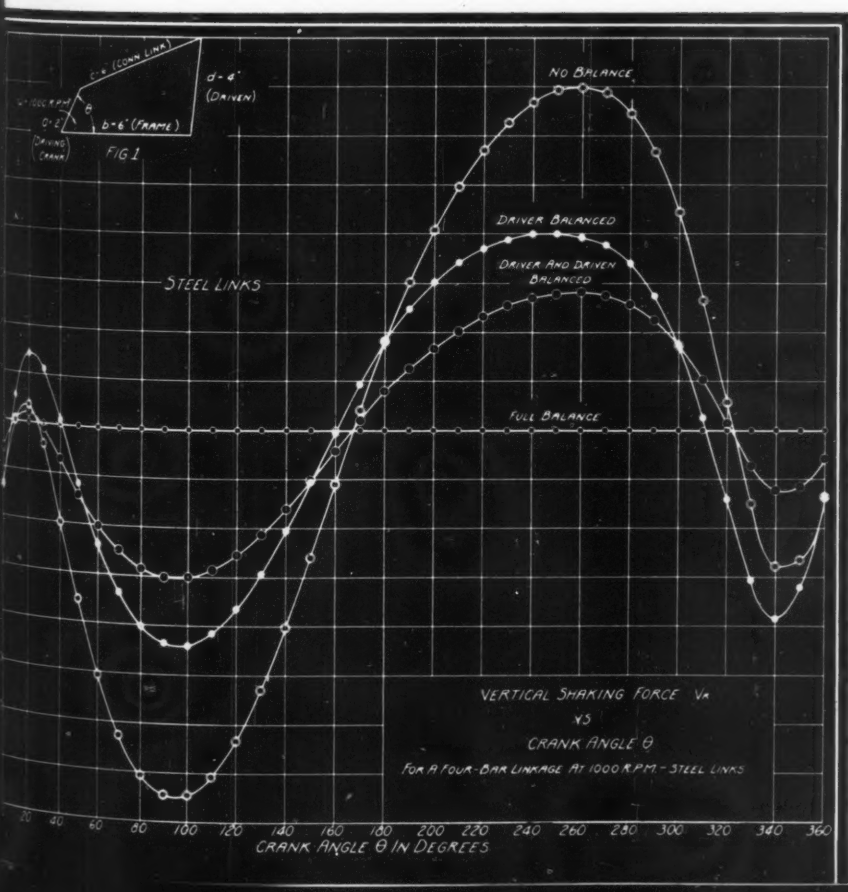
3. Weight of the connecting link and the polar moments of inertia of the cranks should be held to a minimum. For constant speed cranks this is unimportant, but a flywheel should be used to reduce angular speed fluctuations.

Application of Balancing Procedure



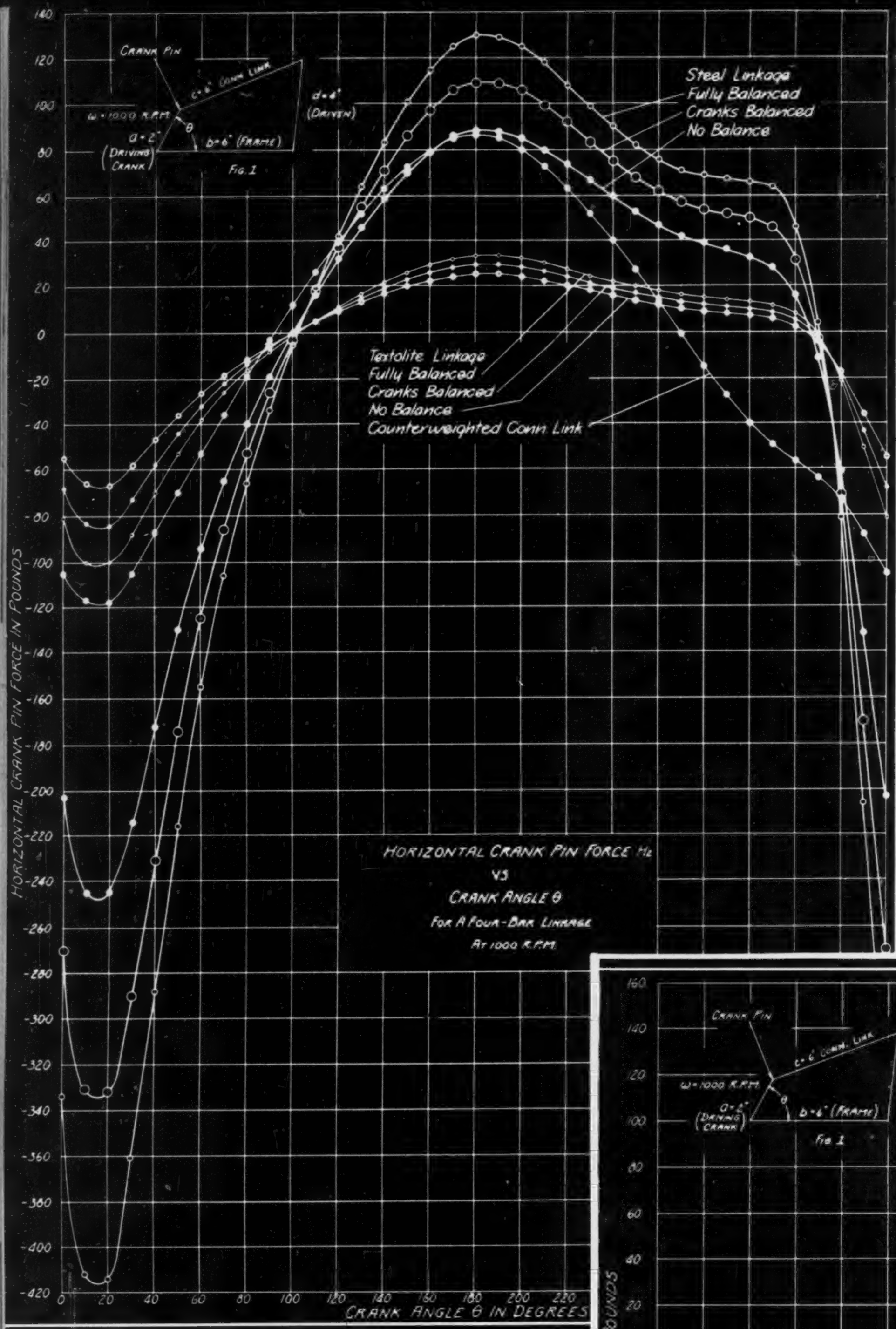
SUBSTANTIATION of formulas and procedure for balancing linkages, as detailed in the Data Sheets, pages 73 and 74, was obtained by tests on a model. In this model the driving crank, *a*, was made 2 inches; driven crank, *d*, 4 inches; connecting link, *c*, 6 inches and distance between fixed bearings, *b*, 6 inches. Both steel and textolite cranks and links were used. The following curves, plotted by means of the formulas in the Data Sheets, agree accurately with the experimental results.

Balancing conditions investigated were as follows: (1) All links unbalanced, (2) driving crank balanced, (3) both cranks balanced, (4) both cranks balanced, and unbalance of connecting link balanced on driving and driven cranks.



VERTICAL and horizontal shaking forces, respectively, are defined as the algebraic sum of the forces acting on the frame at the fixed bearings. These sums are zero for a linkage fully balanced in accordance with the procedure outlined in the Data Sheets. However, in the case of unbalanced or partially balanced linkages, shaking forces are materially reduced by use of links of lighter material than steel as, for example, textolite.

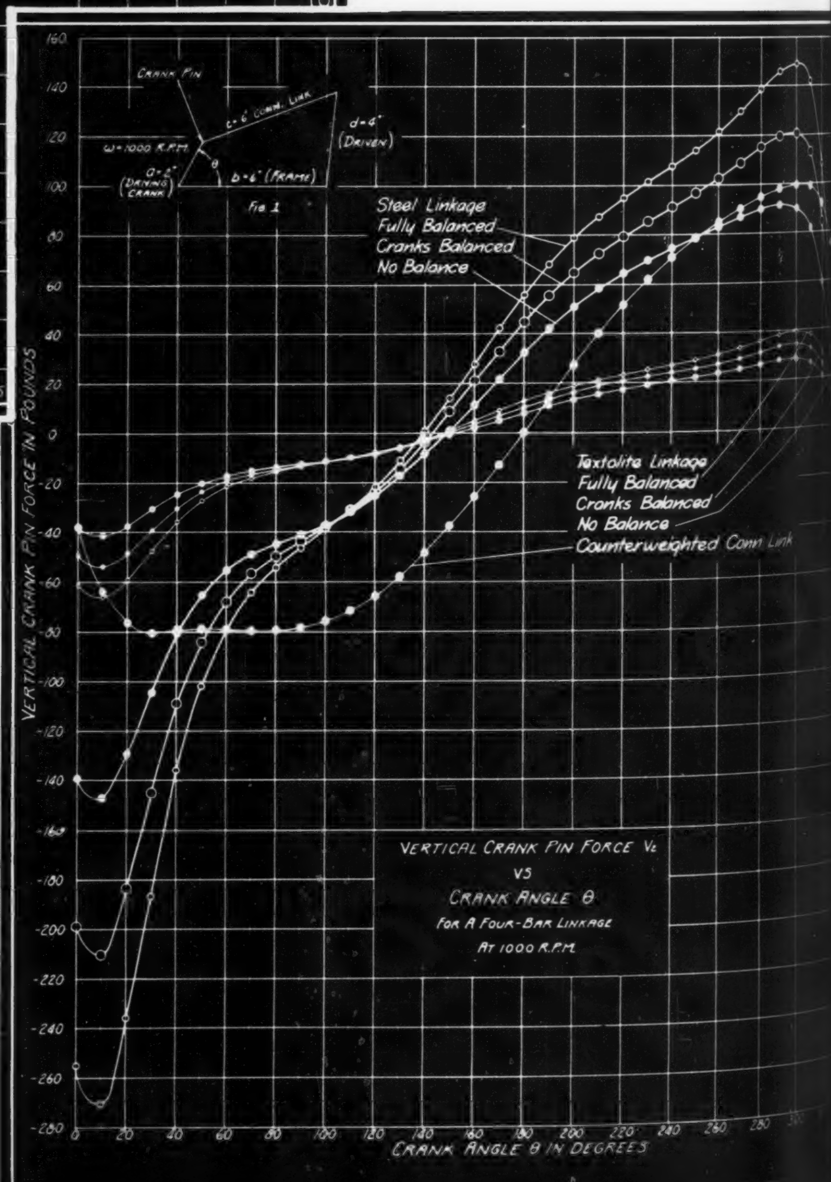
Marked reduction in shaking force obtained by merely balancing the driving crank provides the possibility of improving, with a minimum of redesign, any linkage without ex-



cessively increasing bearing loads as might occur if shaking forces were entirely eliminated by perfectly balancing the linkage.

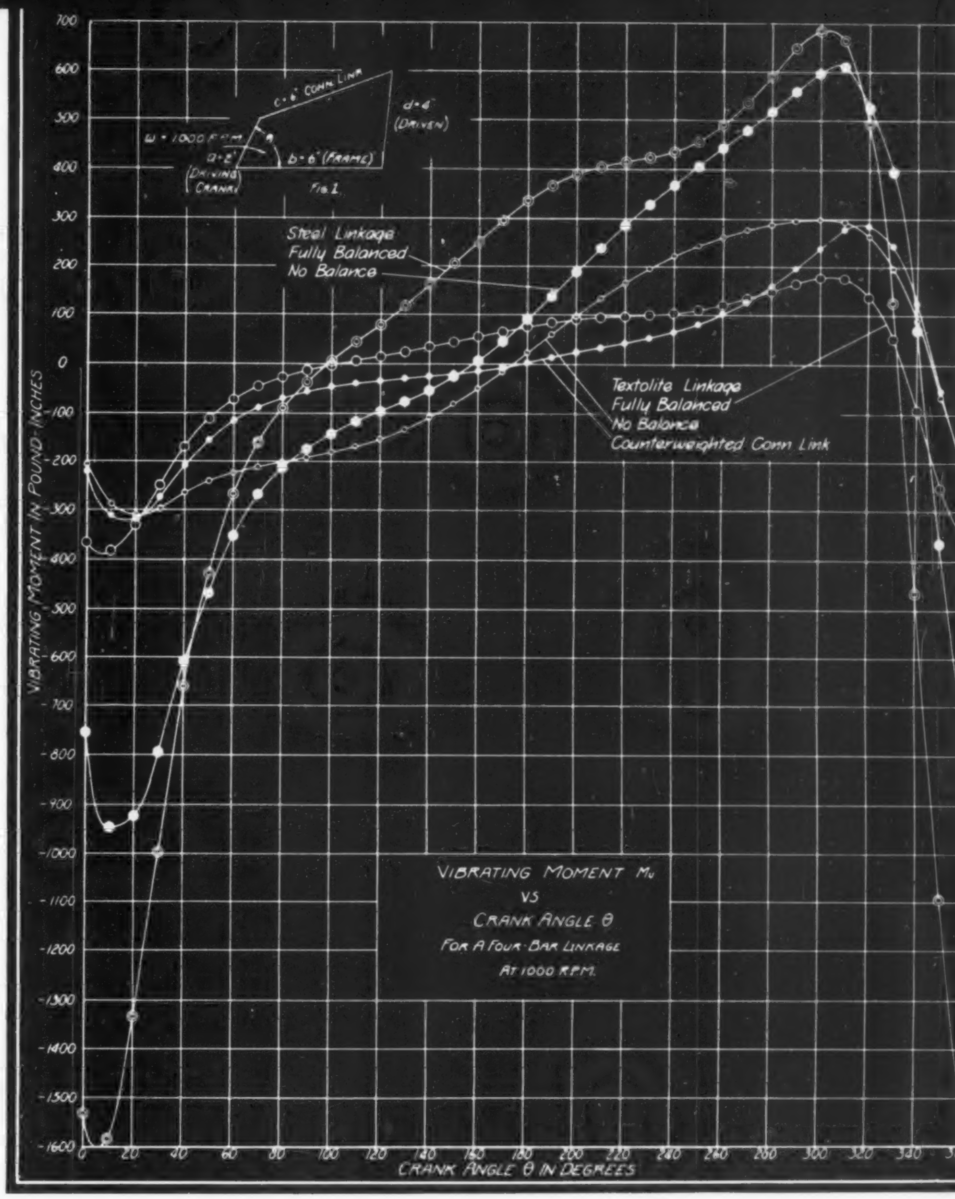
VERTICAL and horizontal crank-pin forces, respectively, are materially increased by balancing out the shaking forces. However, this increase becomes almost negligible if it is possible to use linkage components of a light material such as textolite. Higher pin loads of balanced linkages necessitate careful attention to bearing specification and installation.

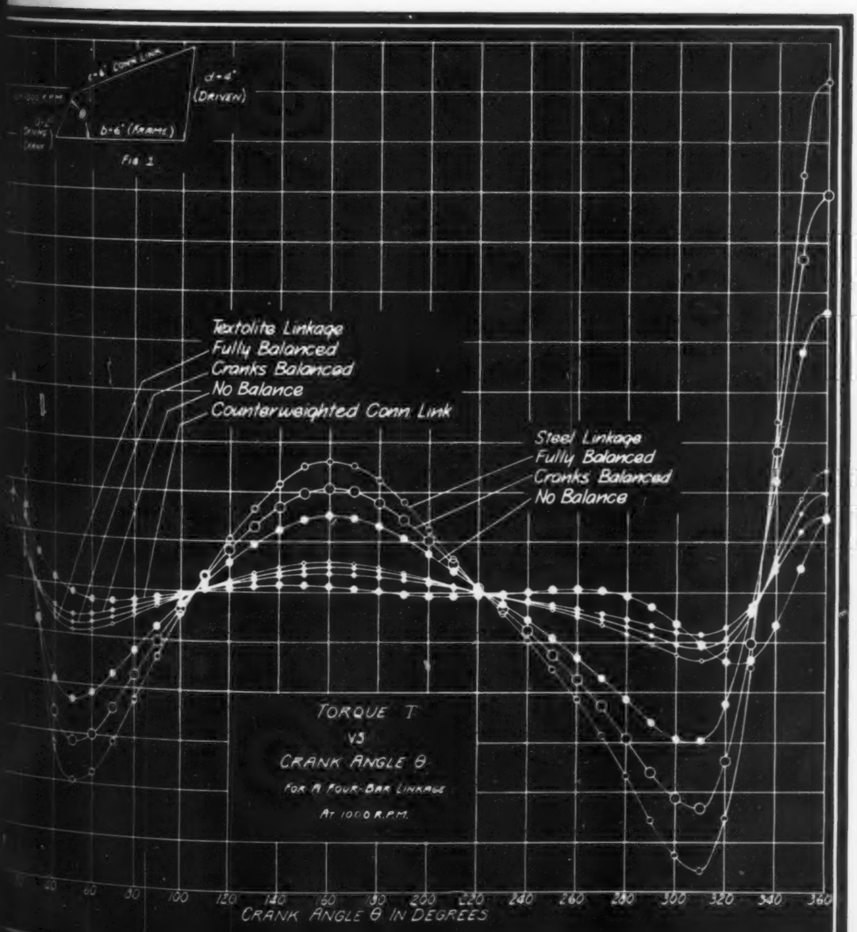
Of the two methods of balancing the effect of the connecting link, i.e. (1) making its center of gravity lie on the center of the driving pin, and (2) balancing half of the connecting link on each crank, the latter results in substantially lower pin forces. With textolite for the connecting link and driven crank, pin forces are increased by the second method from +28 and −41 pounds to +38 and −66 pounds in the vertical plane and from +25 and −68 pounds to +33 and −100 pounds in the horizontal plane. By the first method these forces are increased to +100 and −80 pounds in the vertical plane and to +86 and −120 pounds in the horizontal plane.



ALSO increased by the elimination of the shaking forces is the vibrating moment caused by equal but opposite forces at the two frame bearings, 1 and 4, tending to rock the frame. Again, lighter links reduce the effect. Mass and proportions of the supporting frame are factors to be considered in reducing this influence. Center about which moment is taken is midway of the frame bearings.

Balancingw eights, B_a and B_d referred to in Item 2 of the foregoing procedure should be applied to extensions of the cranks on the opposite side of the stationary frame bearings from the crankpin bearing. Center of gravity of these weights should be made to lie as close as possible to the centerlines of the cranks. Since a moment rather than a mass must be introduced to effect balancing, the weights to be added should be as large as possible in order that the moment arms may be a minimum thereby minimizing the polar moment of inertia.



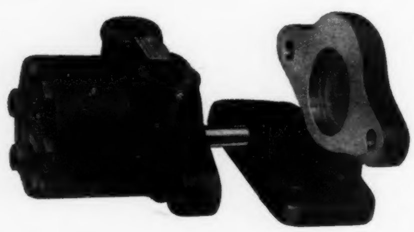


UNLESS flywheel effect of driving crank is adequate, torque fluctuations large enough to cause troublesome vibrations are possible. Because of increased mass of the system, these are also increased by balancing of the linkage. Conversely, steps which can be taken in the direction of reducing the mass of the system will reduce the effect.

New PARTS AND MATERIALS

Rotary Pump Has Helical Gears

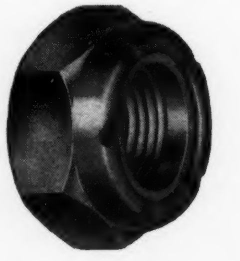
FOR supplying oil or coolant where small capacities are desired, Brown & Sharpe Mfg. Co., Providence, R. I., has introduced its No. 00 rotary geared pump with helical gears providing smooth, quiet operation at speeds adapting the pump to direct motor drive. The pump can be used with pressures up to 100 pounds per square inch, and is suitable for lubrication, for general purposes and for hydraulic installations. Gears are heat-treated steel; shafts are casehardened. Housing cap and



stand are cast iron. Shafts turn directly in the cap and end stand, no separate bearings being used. To prevent leakage, an oil seal is used providing for free turning and requiring less power than a packed gland. The pump is made to run in one direction only, either clockwise or counterclockwise. Capacity in gallons per hour at 0 pounds pressure, at 1725 revolutions per minute, is 30.

Nut Is Self-Locking

FOR general application to light and medium stress fastenings and for use on shear bolts where high proportion of the stress is lateral, an improved line of thin hex nuts is announced by Elastic Stop Nut Corp., 2332 Vauxhall road, Union, N. J. Having approximately 40 per cent of the strength of standard-height hex nuts, the nuts have been developed to meet the demand in industry for a self-locking fastening



which offers saving in space requirements and weight. As in the standard-height line of nuts, the self-locking action is accomplished by means of a vulcanized fiber collar built into the head of the nut. This material being tough and bone-like, resists the entry of the bolt, thus forcing the nut outward and taking up all thread play. The resilient, nonmetallic fiber does not deteriorate under vibration and continues to hold threads of nut and bolt in a constant pressure-contact. Available in steel, brass and aluminum in a complete range of standard sizes, both coarse and fine thread, the nuts are also approved for use on aircraft.

Motor Is Explosion-Proof

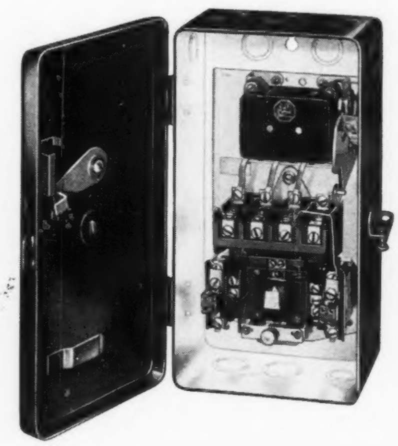
CONSTRUCTED to meet the specifications and carry the label of Underwriters Laboratories Inc., for Class I, Group D installations, a new motor

has been announced by Century Electric Co., St. Louis. This explosion-proof motor for use where the surrounding atmosphere is charged with explosive quantities of acetone, alcohols, gasoline, lacquer, naphtha, natural gas, petroleum, or solvent vapor, is constructed so it will resist the pressure of an explosion of any of these gasses should one occur inside the motor and not cause an explosion in a similar atmosphere outside the motor.

Combination Starter Introduced

AMONG several new features of a new line of combination starters recently developed by Allen-Bradley Co., 1311 South First street, Milwaukee, are a unique handle-locking arrangement and a compact, high-capacity disconnect switch. The starters save wiring, insure greater safety to operator and result in compact installation by combining in the same enclosure a magnetic switch with a

hand disconnect switch. The controls are available in four sizes, in a variety of enclosures, and with or without fuse clips. Ratings range from 2 horsepower, 220-440-550 volts for the size 0 starter, to 30 horsepower, 220 volts, 50 horsepower, 440-550 volts



for the size 3 starter. All sizes have an interrupting capacity of at least ten times the maximum horsepower rating. Locked rotor currents are broken by either the solenoid switch or the disconnect switch which is located in the front. Space is thus saved for a side-operated lever which has three positions: "Off," "Open," and ""On." The cover cannot be opened unless the disconnect switch lever is indicated as "Open."

Industrial Circuit Breakers

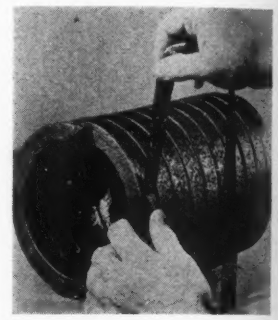
FOR use on 230 volt alternating current systems, the Square D Co., 6060 Rivard street, Detroit, has announced a new industrial multibreaker in both two and three pole form in capacities of from 15 to 100 amperes. Enclosure is dustproof with a felt gasket between box and cover. Including the die cast external handle, the operating mechanism is mounted on the cover of the box so that when the cover is removed the entire interior of the box around the breaker is left free for wiring.

On the cover assembly, a quick make and break mechanism is also incorporated. Provision is made to indicate a tripped condition of the breaker by a white target signal which appears behind a glass window. The 50-ampere multibreaker measures 11⅛ x 6⅝ inches and the 100 ampere size, 13⅛ x 9⅝ inches. The breaker unit used is of the common trip type so that an overload on any pole will

cause all poles to trip simultaneously. This eliminates possibility of the breaker causing single phasing of polyphase circuits. A thermal element affords a time lag on momentary overloads while a magnetic trip feature causes instantaneous tripping on heavy short circuits. Both box and cover of the breaker are made of black enameled sheet steel.

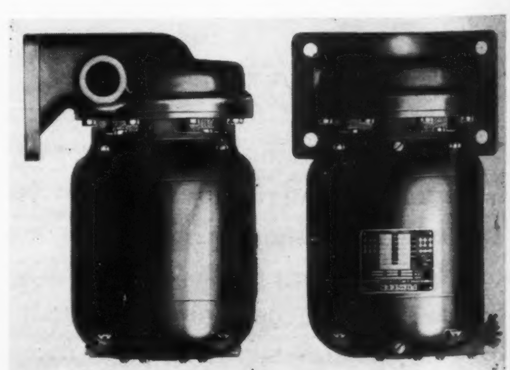
Low Pressure Plastic Packing

USED for packing centrifugal pumps, low pressure steam rods, valve stems, rotary pumps and expansion joints, a new plastic packing with reinforced tape vulcanized to outer surface, has been announced by The Crane Packing Co., 1800 Cuyler avenue, Chicago. This development is known as the Super-Seal plastic packing. Being extremely pliable, the packing can be bent around small diameter rods and shafts without fracturing or cracking, or formed into small inside diameter rings without distortion. It is a dry-graphitized plastic packing produced from long-fiber asbestos, antifrictional metal particles and special binders, and is available in six styles in sizes from ¼ to 1-inch, graduated by sixteenths. It is furnished in coils, standard length spirals or in die-formed rings or sets to stuffing-box dimensions.



Pump for Low Water Levels

MODEL "VBA", a new coolant pump announced by Pioneer Engineering & Mfg. Co., Detroit, is distinguished by its compactness which makes possible the placement of the coolant pump under conditions beyond the range of other models. The new pump is capable of functioning with an extremely low water level. Positive prime is



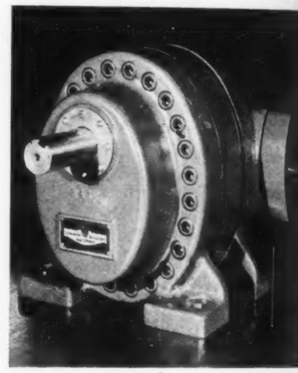
(*Continued on Page* 88)

(Continued from Page 82)

assured even with the reduced height of coolant in the supply tank. The pump, which can readily be taken apart and serviced, may be mounted on the outside of the machine or tank below fluid level. Mountings are made with two outlets, either right or left-hand. Motors have also been improved in contour design and by the absence of inlet piping, lending a note of neatness to the appearance of the machine to which the pump is mounted. The same improved design is also carried throughout the entire line of the company's pumps.

Gear Type Hydraulic Pump

ANNOUNCEMENT of a new, larger size, gear-type, hydraulic pressure generator has been made by Hydro-Power Systems Inc., Mount Gilead, Ohio. Model "G"-60 gear pump, of constant delivery type, has a delivery of 60 gallons of oil per minute at 1000 pounds per square inch line pressure. It has heavy-duty construction with flange connections. Close tolerances are maintained between working parts so that pump output efficiency is high. Precision spur gears of narrow width and large pitch diameter are used, reducing both hydraulic load on gears and distance between gear shaft bearings so that shaft deflection is minimized. Provision is made for long bearing life and accurate positioning of pump gears by mounting gear shafts on precision tapered roller bearings. These gear pumps are used for various applications such as prime movers for hydraulically-operated machines requiring medium pressures; as drives for auxiliary units, etc. Model "G" pump is recommended for operation at 1000 pounds per square inch on such applications where peak pressures are "flash" or momentary; for operating up to 500 pounds per square inch where peak pressures are maintained or held for any appreciable time.

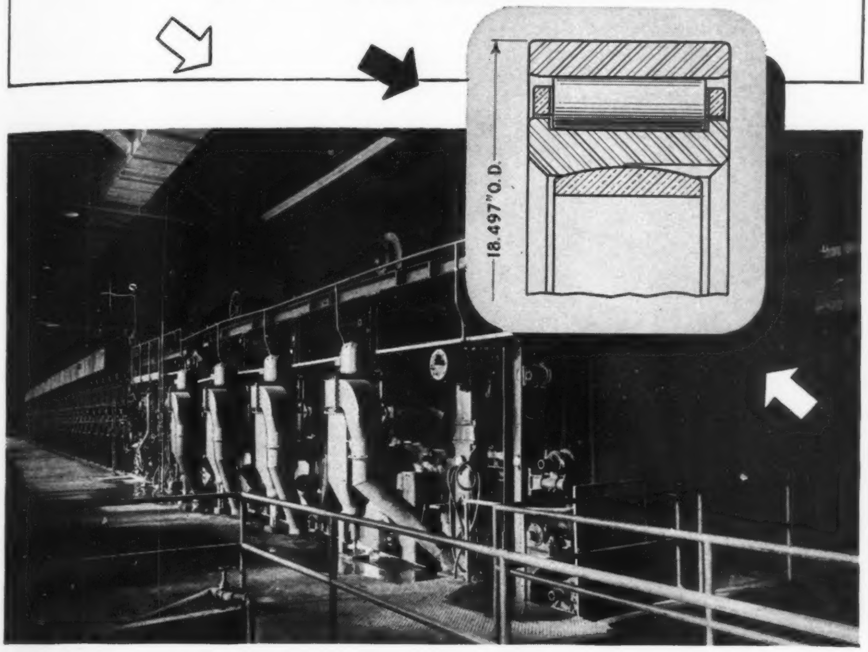


Starter for Mining Service

FOR mining equipment such as conveyors, pumps, fans, etc., remote and automatic control has been provided in the new d-c combination starter announced by Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Automatic starting and acceleration are accomplished from a built-in or remotely mounted pushbutton. A line disconnecting switch, low voltage protection, instantaneous overload protection, and safety interlocks to prevent

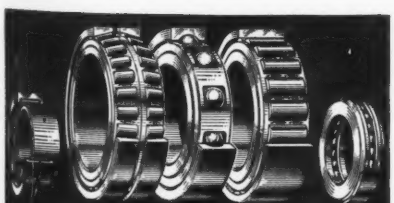
IN THE NEWS

WITH BANTAM BEARINGS

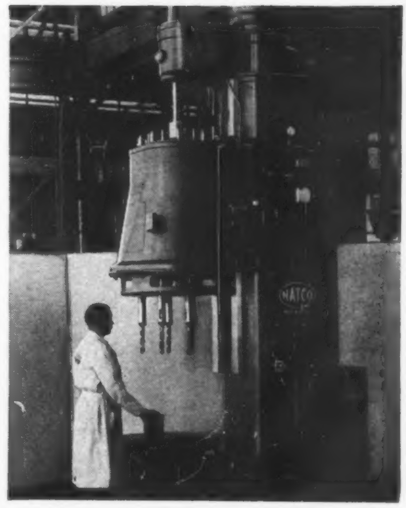


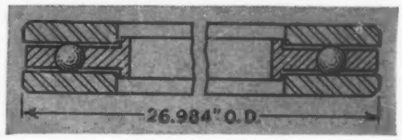
BUILT FOR FAST PRODUCTION, this paper-board machine at the plant of Alton Box Board Company turns out a continuous sheet of strawboard, .009" thick, at the rate of 520 feet a minute. Rolls furnished by Beloit Iron Works are equipped with large, heavy-duty Bantam Bearings. Supplying specially designed bearings for many applications in paper, steel, and rubber mills, and other industries where heavy loads and severe service are encountered is a major part of Bantam's service.

POWER ECONOMY is the design keynote of Oliver Farm Equipment Company's Row Crop 60 Tractor—built for heavy duty farm service. An important factor in its economy of operation is the use of Bantam Needle Rollers in the cluster gear transmission to provide efficient anti-friction operation in extremely limited space.



EVERY MAJOR TYPE OF ANTI-FRICTION BEARING is included in Bantam's line—tapered roller, straight roller, needle, and ball. Bantam engineers, with their broad background of experience in bearing design and application, recommend the type that best meets *your* requirements—or design special bearings for unusual conditions. If you have a difficult bearing problem, TURN TO BANTAM.





THIS HEAVY-DUTY DRILLING MACHINE of the multi-station type, built by The National Automatic Tool Co., is provided with a 36-inch diameter automatic indexing table. Table rotates on specially designed Bantam Ball Thrust Bearing, measuring 26 63/64" O.D., with load capacity of 8,000 pounds at 10 RPM, 30,000 pounds stationary.

HIGH OSCILLATING LOADS on the saddles and pitmans of The National Supply Company's pumping units are easily handled with Bantam Quill Bearings—the compact anti-friction bearings that combine high capacity with low cost and small size. For additional information on the Quill Bearing, write for Bulletin B-104.

BANTAM BEARINGS

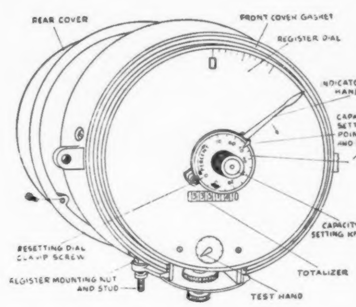
STRAIGHT ROLLER · TAPERED ROLLER · NEEDLE · BALL

BANTAM BEARINGS CORPORATION • SOUTH BEND • INDIANA

opening the door unless switch is off, are some of the safety features included. When disconnecting switch is closed and start button pushed, the line contactor closes, starting the motor with resistance in the circuit. The coil of accelerating contactor is connected across motor armature. As the motor accelerates, the voltage across the armature increases and causes the accelerating contactor to close at the proper time. This short circuits the resistor and connects motor across the line. A small adjustable field rheostat may be added when an accurate definite motor speed is desired, providing up to 15 per cent increase in speed.

Recycling Control Is Automatic

FOR use with meters to initiate regeneration of automatic processes, a recycling control is being supplied by The Permutit Co., 330 West Forty-second street, New York. The control is actuated by a synchronous electric motor. Mercury switches are employed throughout to ensure dustproof, cor-



rosion-resistant contacts. There are no electrical current carrying parts on the dial. When predetermined amount of water has passed through the meter, contact is made to initiate automatic cycle. The meter control then resets itself and is ready automatically to measure another predetermined quantity of water. By means of an easily accessible dial visible through a glass face, adjustment of the quantity of water to pass through the meter is accomplished.

Split and Solid Journal Bearings

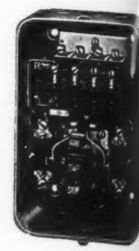
SOLID and split journal bearings have been improved and can now be furnished promptly by The Jeffrey Mfg. Co., Columbus, O. Of accurate

dimensions with modern rounded lines and smooth gun metal finish, these precision-made bearings

have machined bases and faced ends. Height to center line of shaft is rigidly maintained. The babbitted bores are broached to a smooth hard surface and require no wearing-in. Both the solid and split types are tapped for grease cups or pressure fittings and an ample storage groove in the top provides proper distribution of lubricant. The split bearing, in addition, has feeder grooves on each side.

Solenoid Starter Is Magnetic

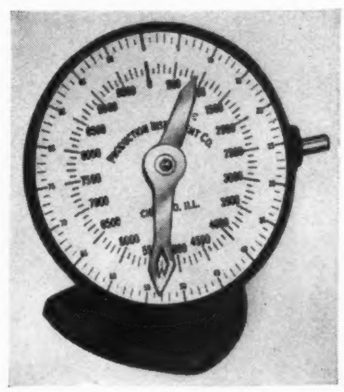
MAGNETIC solenoid starters and contactors, in sizes "0" and "1," have been announced by The Arrow-Hart & Hegeman Electric Co., Hartford, Conn. Separate bakelite units for each pole confine the arc and exclude dust. All connections are made from the front, and coils with clearly visible ratings are easily changed. Separate electrical interlock switch, enclosed in bakelite, is supplied normally open or closed. Positive quick make and break switching mechanism in overload relay



is insulated from bimetal and heater. Overload relays are regularly supplied for manual reset, but can be changed for automatic reset if desired. Reset button is independent of the cover. The starters and contactors are for control of motors up to 7½ horsepower single or polyphase, 110-550 volts, alternating current.

High-Speed Counter Improved

AN IMPROVED high-speed coil winding counter has been introduced by Production Instrument Co., 702-08 West Jackson boulevard, Chicago. Designed for direct connection to the motor shaft or for operation through a flexible shaft, the counter,



in actual service, counts at speeds as high as 9000 turns per minute. A single motion resets the counter to zero, and a unique pointer design gives instant visual check of reset. It also deducts turns taken off the coil in case of overrun, permitting accurate

A crystal-clear synthetic solid age-proofs this tracing paper
MINERAL OIL. Most tracing papers are treated with some kind of oil. Mineral oil is physically unstable, tends to "drift", never dries completely. Papers treated with mineral oil pick up dust, lose transparency with age.
VEGETABLE OIL, chemically unstable, oxidizes easily. Papers treated with vegetable oil become rancid and brittle, turn yellow and opaque with age.
ALBANITE is a crystal-clear synthetic solid, free from oil and wax, physically and chemically inert. Because of this new stabilized transparentizing agent Albanene is unaffected by harsh climates—will not oxidize with age, become brittle or lose transparency.
No oil, no wax—but a remarkable new transparentizing agent developed in the K&E laboratories—produces this truly permanent tracing paper! ALBANENE is made of 100% long fiber pure white rags—treated with *ALBANITE*—a new crystal-clear synthetic solid, physically and chemically inert. ALBANENE will not oxidize, become brittle or lose transparency with age.
Equally important, ALBANENE has a fine hard "tooth" that takes ink or pencil beautifully and erases with ease ...a high degree of transparency that makes tracing simple, produces strong sharp blueprints... extra strength to stand up under constant corrections, filing and rough handling. ALBANENE has *all* the working qualities you've always wanted—and *it will retain all these characteristics indefinitely.*
Try ALBANENE yourself on your own drawing board. Ask your K&E dealer, or write us, for an illustrated brochure and a generous working sample.
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THE STABILIZED TRACING PAPER

winding at high speed. The counter, equipped with hobbed gears and Oilite main bearing, has large legible figures reading to 10,000 turns. Pointers have specially designed friction hubs which afford large bearing area for dependability and minimum wear. An adjustable base provides for easy alignment with motor.

Induction Motor Is Splashproof

TO SUPPLEMENT its recently announced Tri-Clad polyphase and single-phase open general-purpose motors, the General Electric Co., Schenectady, N. Y., has introduced a line of Tri-Clad splashproof, ball-bearing, polyphase induction motors in sizes from 1 to 15 horsepower. This motor is es-

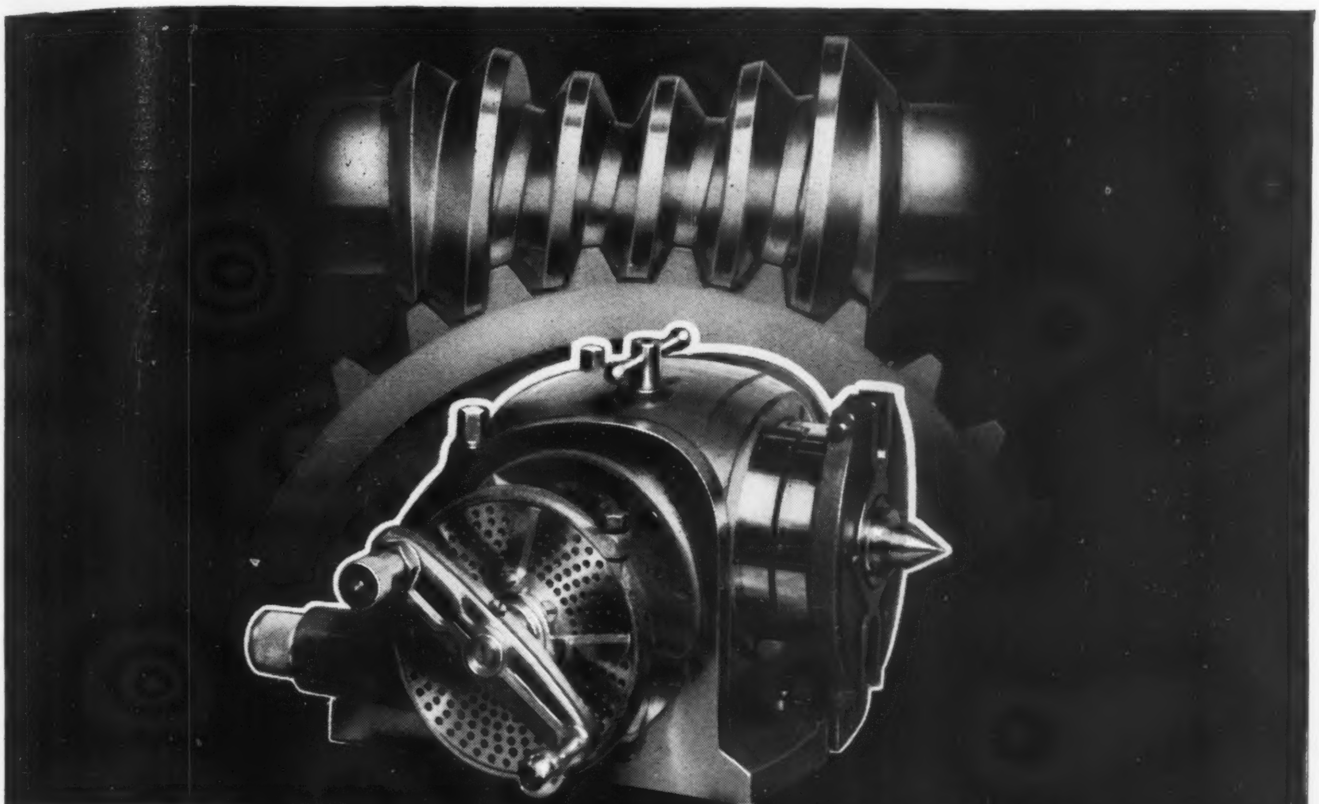
pecially designed to meet needs of applications where splashing water and other liquids are present, such as on machines for dairies, breweries, paper mills, food industries, packing plants, laboratories and similar applications. Features of the motor include cleanlined appearance, triple protection—against physical damage, electrical breakdown, and operating wear and tear—sturdy cast-iron stator frame and end shields, resisting rust and corrosion, wire windings highly resistant to moisture, and thoroughly protected ball bearings.

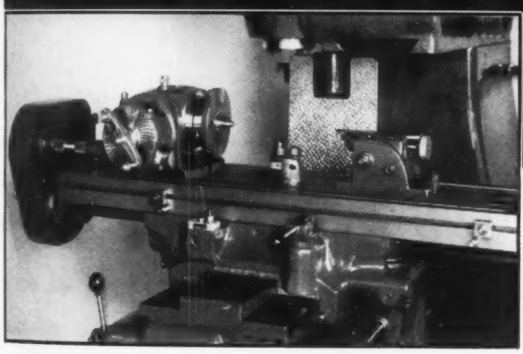
High Strength Malleable Offered

ANOTHER new material, Belectromal, has been made available by the Belle City Malleable Iron Co. and Racine Steel Castings Co., both of Racine, Wis. Melted and alloyed in an electric furnace, the metal is a malleable iron used for parts requiring high strength with an ability to withstand shock and rough handling in the machine shop. The company now supplies five types of ferrous castings and three of the malleable.

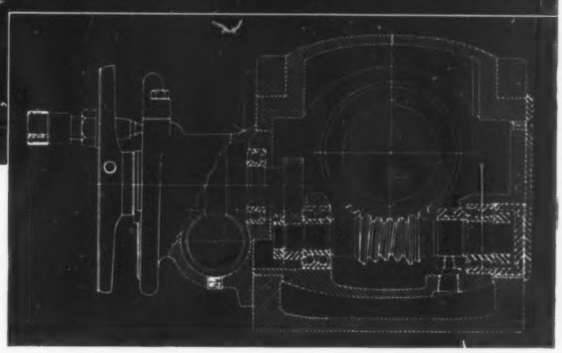
Controllers Are Motor Operated

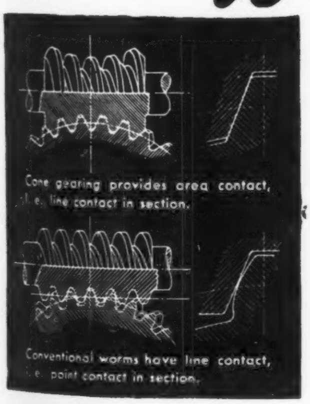
NEW motor-operated secondary controllers for use with polyphase wound-rotor motor on fan, pump and similar drives, have been announced by Westinghouse Electric & Mfg. Co. Either 13 or 20 balanced points of control are pro-





Above: Van Norman's new Cone-driven dividing head.
At left: Mounted on a Van Norman Universal miller.
At right: The Cone-Drive layout used in the Van Norman head.

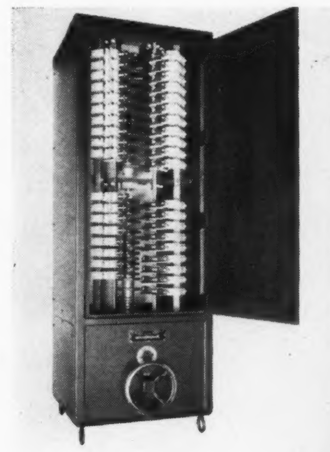




Cone gearing provides area contact, i. e. line contact in section.

Conventional worms have line contact, i. e. point contact in section.

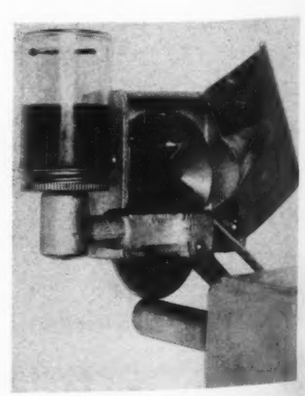
vided on the new SC controller by varying the external resistance in the motor secondary winding. Enclosed in a self-supporting steel cabinet, the unit has cam-actuated contactors arranged for sequential operation in pairs from a common motor-driven camshaft. Individual cams give quick



"make" and "break" contact action. Drive mechanism consists of a pilot motor, a gear reduction unit, and a geneva gear for angular movement of camshaft. Step-by-step action is assured by a gear and pinion arrangement to move controller one full position or point for each gear revolution. Overtravel protection is provided by auxiliary cam-actuated switches. The motor has such optional features as remote position indication, follow-up attachment and time delay operation, and can be obtained for 115 or 230 volts direct current and 110, 220 or 440 volts alternating current.

Automatic Metering Lubricator

SUPPLYING the right amount of lubrication at the right time for vacuum pumps, a new automatic mechanical metering lubricator has been developed by Leiman Bros. Inc., 101-P-7 Christie street, Newark, N. J. Suitable for use on pumps and for other difficult lubricating problems, the company is now offering it for use on any machine or device where a moving part is to be lubricated. Friction contact of the rotating attachment of the lubricator with the surface of any shaft furnishes the means of starting and stopping the flow of the oil. Actual feeding is performed by a device which may



be regulated and set to feed from one drop in ten minutes to ten drops in one minute. It does not rely on gravity feed for the oil to reach the surface requiring lubrication; the device parcels out oil at set speed, and when machine stops the oil ceases. The lubricator has a visible oil container.

Meter Attached to Analyzer

TO INSURE greater accuracy in making measurements with its new surface analyzer, the Brush Development Co., Cleveland, has introduced a new meter attachment which provides a visual

indication of the root-mean-square average height of irregularities expressed in microinches. With a flip of a switch the meter will make a permanent and instantaneous record of the actual heights and depths of the irregularities in microinches.

New Converter Announced

MORE compact, modern and convenient the new converter, announced by EICOR, 1060 West Adams street, Chicago, is intended for portable or stationary use. It converts direct to alternating

current for amplifiers, projectors, phonographs, radio receivers, transmitters, medical equipment, musical instruments and other applications. Supplied with or without filter the converter is available for 6, 12, 32, 115, 230 volts or other standard DC input—and has standard AC output. It is equipped with ball bearings.

MEN Of Machines

APPOINTMENT of Dwight Richards as chief engineer of the Railroad division of The Buda Co., places in charge of the development of new equipment one who is ably suited to handle responsibilities. Since graduating in mechanical engineering from Ohio State, Mr. Richards has followed his profession closely and gained valuable experience. Before joining the Buda company he was chief engineer of the English company of Sullivan Machinery Co. (in England) in charge of the design and manufacture of coal mining machinery, hoists, drills, rock-loading equipment and other machinery built by the company for England, Europe and South Africa. He was with the organization for seven years but due to the war he returned to the United States in July, 1940. Prior to his connection with Sullivan, Mr. Richards was chief engineer for the Myers-Whaley Co. He has also been associated with Insley Mfg. Co., and Marion Steam Shovel Co., in engineering capacities.

ELECTION of Walter P. Schmitter, chief engineer of The Falk Corp., Milwaukee, as president of the American Gear Manufacturers' association has recently been announced. Mr. Schmitter has been active in the formulation of practices of the association and has contributed widely to technical gear literature in the form of numerous papers and articles, as well as addressing many important groups. Holding num-

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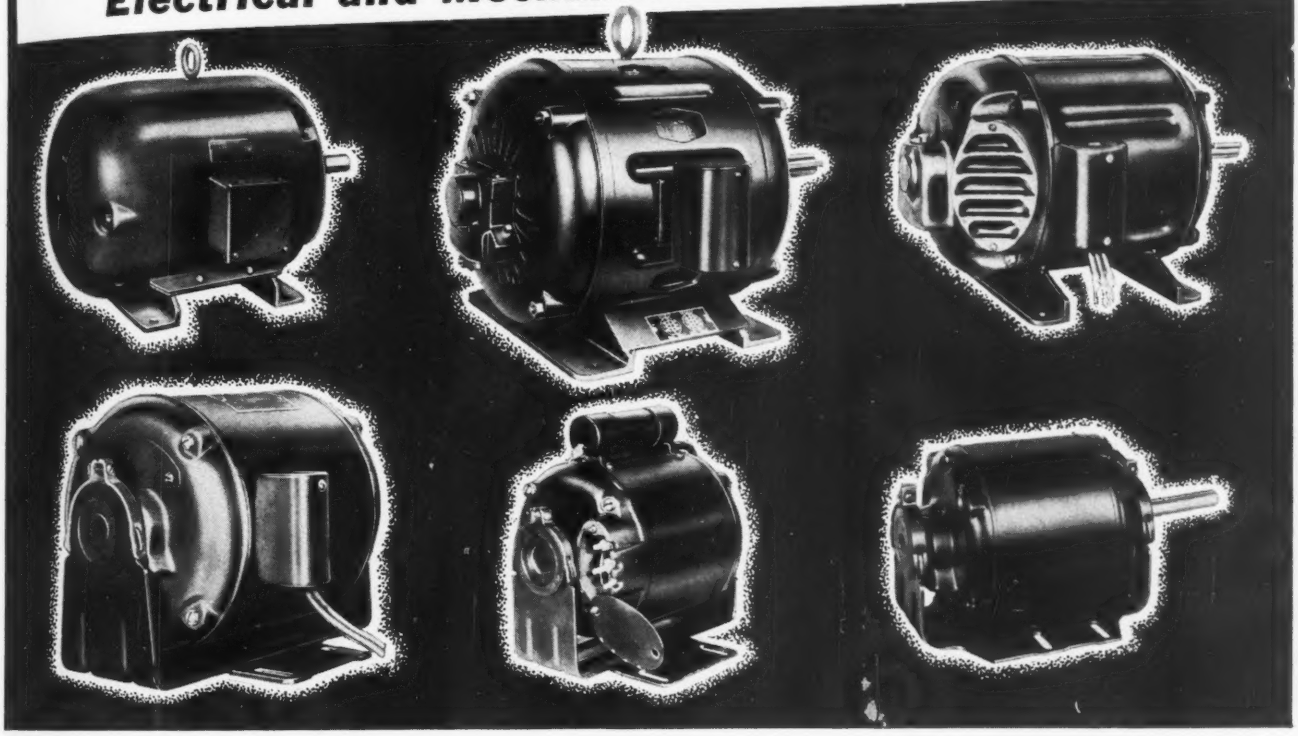
. . . a Wide Range of Types and Sizes with varied Electrical and Mechanical Characteristics . . . .



A good motor is not enough—it must be the right motor for the job. In order to properly select a motor, the following points should be considered.

Load Cycle . . . What maximum and minimum horsepower is involved, and what is the probable duration of each?

What are the maximum starting torque requirements?

Is the duty cycle continuous or intermittent, and what method of control and overload protection is contemplated?

Power Supply . . . A. C. or D. C., and frequency if A. C. . . . Voltage . . . Phase . . . Special starting current limitations, if any, imposed by the power supplier.

Speed Characteristics . . . Single constant speed . . . Variable or multispeed.

Mechanical Construction . . . Is open type motor acceptable . . . or should Splash proof, Totally-enclosed fan-cooled, or Explosion-proof motor be used?

A Wagner, factory-trained, sales-engineer is ready to assist you in selecting the proper motor for your specific requirements.

WRITE FOR BULLETINS MU-177, MU-179 and MU-182

Diversified Types: It's easy to select a WAGNER motor that exactly fits the job because there is such a wide range of types and sizes from which to choose, each one designed and built for a particular kind of service.

Modern Design: Wagner motors are neat and compact, up-to-the-minute in design, and harmonize with the most modern machinery and equipment.

Dependable Performance: Wagner motors are well engineered and sturdily constructed, making them absolutely dependable even when operating under the most severe service conditions. A fifty-year reputation for satisfactory performance is your assurance that the best buy in motors is Wagner.

Quick Service: With branch offices and warehouses in 25 major cities, and a centrally located plant at St. Louis, Missouri, the Wagner organization is equipped to render immediate service anywhere in the country.

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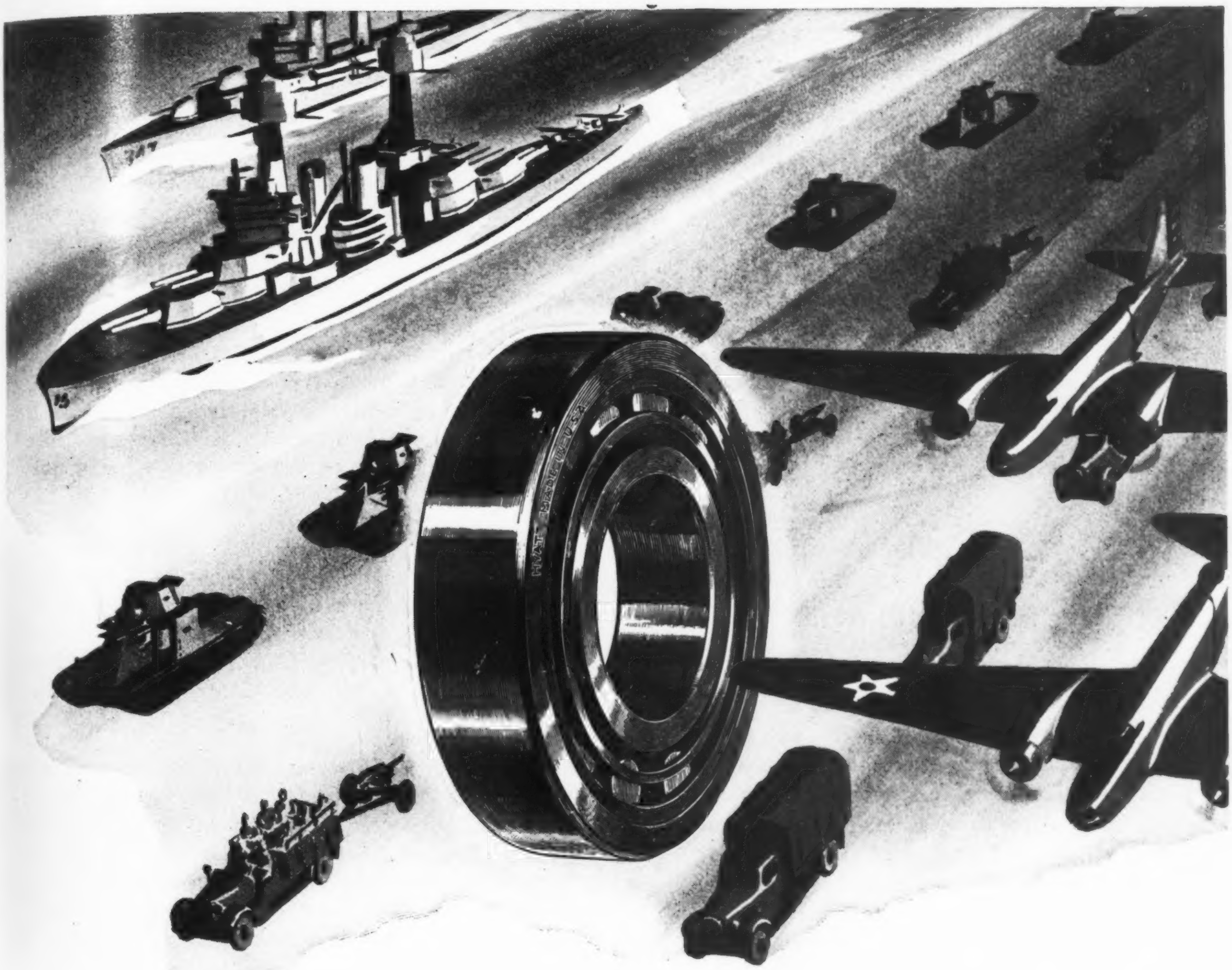
MOTORS • TRANSFORMERS • FANS • BRAKES

erous major patents pertaining to gears and power transmission devices, he is well versed in this subject. Mr. Schmitter received his early training at Dickinson Technical and Industrial school and the Brooklyn Polytechnic institute. For short periods later he was employed with the American Machine & Foundry Co., Cutler-Hammer Mfg. Co., the Allis-Chalmers Mfg. Co., and spent some time in consulting practice in Milwaukee in special machine design. He became connected with Falk in 1923. At the present time, in addition to his new post, Mr. Schmitter is a member of the executive committee of the A. G. M. A., chairman of the helical and herringbone committee of this association, a member of the Association of Iron and Steel Engineers, and the Army Ordnance association.

AS PRESIDENT of the American Society of Tool Engineers, Frank W. Curtis is in a position to place to good use his experience and knowledge in the engineering field. He is associated with the Van Norman Machine Tool Co., Springfield, Mass., as chief engineer in charge of milling machine design, and is also a member of the executive board of that concern. Graduating from high school in New York, Mr. Curtis devoted his studies to mining engineering and became connected with one of the largest lead and zinc mining companies in the Joplin area, which he managed for some time. He then studied mechanical engineering and went to Detroit where for many years he worked in the capacities of tool and machine designer, production supervisor and efficiency engineer, with concerns such as Ford, Studebaker and Maxwell (now Chrysler). Later he became identified with the design of special machinery and development of production equipment. After working in an editorial capacity with a trade journal he joined Kearney & Trecker Corp., Milwaukee, as research engineer. His next connection was with the Van Norman Machine Tool Co.

Carl A. Cover has been named executive vice president and top ranking official of the Douglas Aircraft Co., directly under Donald W. Douglas, president. To make improved co-ordination possible, necessitated by increased production activities and plant expansion, five new advisory councils have been named, one of which is the new products council under the chairmanship of Mr. Cover. Other members are Arthur E. Raymond, E. F. Burton,



chief designer, and E. H. HEINEMANN, chief engineer of the El Segundo plant. Mr. Cover was formerly vice president and general manager of the company.

R. H. CHADWICK has been made assistant to the manager in charge of engineering, General Electric Co., Schenectady, N. Y. Since enrolling in the test course at Fort Wayne in 1912 he has been a member of the transformer engineering department. C. H. MATSON has been promoted to assistant works manager. He joined G. E. in 1909 as mechanical draftsman, later becoming head of the factory engineering department, in 1919 plant superintendent, and in 1925 assistant to general superintendent. E. J. THOMAS, since 1926 a member of the transformer department, has been named engineer of the specialty transformer department, Fort Wayne.

WILLIAM L. LUDWICK becomes chief engineer of the instrument division of Thomas A. Edison Inc. Formerly he was connected with Farnsworth Television & Radio Corp., where he was chief mechanical engineer in charge of commercial design of television transmitter and high frequency apparatus.

T. E. SHEA, engineering vice president of Electrical Research Products Inc., has been granted a leave of absence to participate in studies for the National Defense Research committee. DR. E. M. HONAN, of the company's Hollywood office, has taken over the direction of motion picture engineering activities in Mr. Shea's absence.

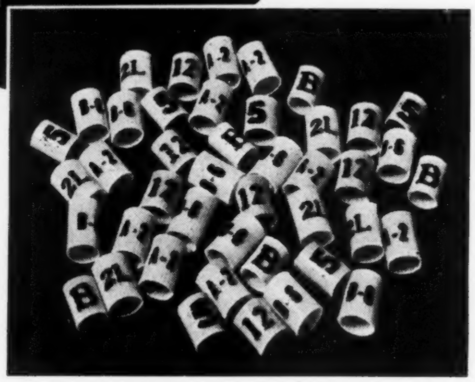
J. M. ROBERTS was re-elected president of the Scientific Apparatus Manufacturers' Association. FREDERICK POST, president, Frederick Post Co., was named to a vacancy on the board of directors created by the retirement of THOMAS LORD, president, C. F. Pease Co.

HARRY L. WILCOX, assistant chief engineer, Electric Controller & Mfg. Co., Cleveland, has been elected president of the Cleveland Engineering Society.

G. L. OSCARSON has been appointed chief application engineer for the Electric Machinery Mfg. Co., with which he has been associated for the last 18 years.

THOMAS J. LITLE JR., formerly chief engineering executive at the Easy Washing Machine Co., has been named director of engineering for the Bendix Home Appliances Inc.

A. W. PARKER is retiring after 54 years' continuous service with the Worthington Pump & Machinery Corp., Harrison, N. J. Well known in engineering circles, Mr. Parker started as a draftsman in 1887 and through many years served as mechanical engineer and product designer. He has made noteworthy contributions to the development of a variety of pumping machinery.

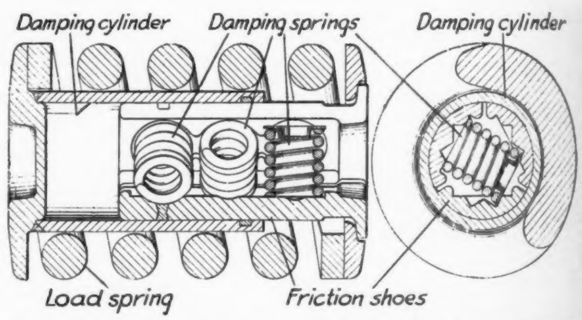


Noteworthy PATENTS

Damps Spring Oscillations

DESIGNED primarily for use in railway trucks, the vibration damping system disclosed in a patent assigned to The Symington-Gould Corp. should find broad application wherever non-harmonic, cushioned supporting means are required. Consisting essentially of a compression load-carrying spring mounted between two flanges, a cylindrical sleeve is welded to the upper flange as illustrated.

Mounted within this sleeve are three friction shoes, the confined outer surfaces of which are



Use of three damping springs provides equalized friction forces for damping harmonic oscillations

practically cylindrical and mate with the inner surface of the sleeve or damping cylinder. The inner surfaces of these shoes are so formed that they provide seats for three helical, compression springs.

Each spring is so arranged that on one end it seats on two shoes and, on the other end, its axis coincides with the center line of the third shoe. Each shoe is thereby loaded radially with respect to the load-carrying spring by an equalized load contributed to by each of the damping springs.

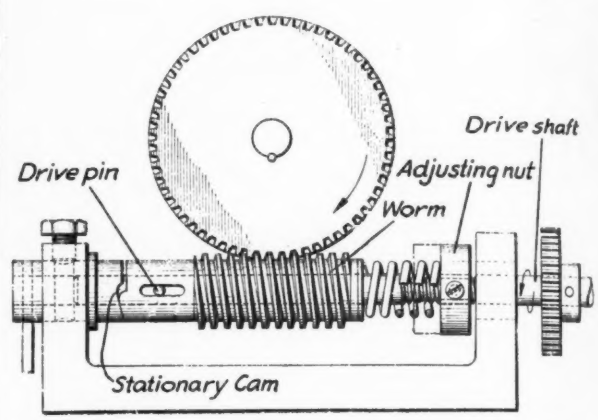
Intermittent Motion Provided

SIMPLIFICATION and economy in the conversion of uniform rotation to intermittent motion is achieved by the mechanism disclosed in a patent assigned to International Business Machines Corp. Ordinarily obtained by geneva mechanisms, cam devices or a ratchet and pawl, such motion is provided in this device by means of constrained axial movement of a worm.

Supported by two end bushings, a driveshaft is driven from any constant speed source. Held in the frame adjacent the left bushing in the illustra-

tion is a stationary cylindrical cam. Coacting with this cam is a contoured surface machined on the sleeve extension of the worm.

In order to provide intermittent rotation of the

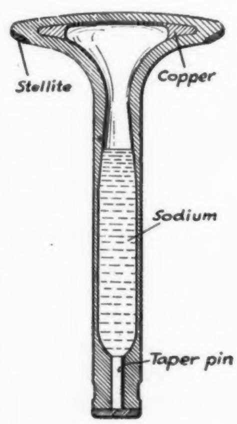


Axial translation of the worm controlled by cylindrical cam affords intermittent rotation

wormwheel, the cam surfaces are so machined that they afford a constant velocity rise for a distance equal to the worm lead in one-half revolution of the driveshaft. During the rise of the cam the spring is compressed, storing energy for driving the wormwheel for the remaining half revolution. Thus for the first half revolution of the worm the axial translation exactly equals the lead and no rotation of the wheel occurs. During the remaining half revolution the worm is rotated by both the axial movement of the worm to the left and by its rotation.

More Efficient Valve Cooling

CONDUCTION of heat away from the radially remote sections of internal combustion engine poppet valves has been greatly expedited by the development of the sodium-cooled valve. In consequence, the life of these units before regrind-



Deposit of copper in the interior periphery of valve head assists in maintenance of low seat temperature

ing has been increased considerably by the provision of more even heat dissipation over the entire surface of the unit.

Extremely high temperatures existing in the neighborhood of the hard-surfaced seating portions

provide an opportunity for further improvement as detailed in a patent assigned to the Eaton Manufacturing Co. After the head has been upset and the stem bored out or swaged, a selected amount of copper in rod form is inserted through the pin hole. The valve is then positioned, head down, in a suitable furnace and its temperature raised an amount necessary to melt the copper. Rotation of the valve about its vertical axis results in the deposition of the molten copper directly behind the seat.

Being an excellent heat conductor the copper removes heat quickly from the seat, thus facilitating its dissipation first to the sodium and thence through the valve stem. After the sodium has been introduced a taper pin is driven into the hole in the stem and the end covered by welding on a cap.

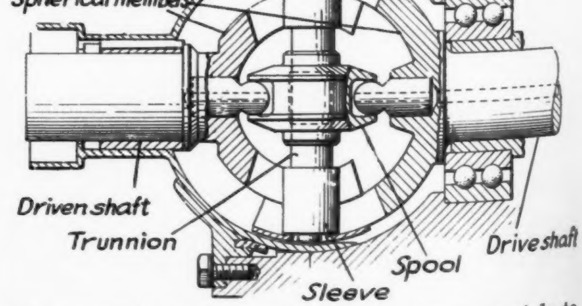
Rotates at Constant Speed

COMPLYING with the basic requirements for constant angular velocity of the driven shaft, this patent, assigned to Universal Products Co. Inc., discloses a universal joint embodying distinct improvements in design. Axis of the universal trunnion always bisects the angle between the driving and driven shafts and the center lines of these shafts intersect at the midpoint of the trunnion.

To simplify the discussion, the right-hand shaft has been designated the driver and the left-hand shaft, the driven. However, it is apparent that the drive may be effected through either. Each shaft, as it enters the universal housing is fitted with a spherical member which co-acts with that of the other forming, for practical purposes, a ball and socket joint.

Each spherical member is provided with diametrically opposed slots in such a manner that they may be brought into register. A trunnion pin with needle bearing sleeve rollers on each end is inserted in the aligned slots, thus preventing relative rotation of the driving and driven spherical members.

Floating in substantially the center of the trunnion is a flanged spool which engages, between its flanges, spherical extensions of the two shafts. Thus as one of the shafts (in this case the driven) is displaced angularly from the center line of the other, the center of such angular rotation is always at the geometric center of the spherical members.



Universal joint utilizes ball and socket principle to afford constant velocity of driven shaft

Designing for Hydraulic Variable Speed

(*Concluded from Page 45*)

the maximum system pressure is to be 1000 lbs./sq.in.:

$$HP_{output} = 10 = \frac{2\pi T_m N_m}{12 \times 33{,}000} \quad \text{(12)}$$

Since the pump will need to operate at higher speed due to an estimated leakage of 10 per cent, we shall express the N_m in terms of pump speed as

$$N_m = .90\, N_p = \frac{.90 \times 2000}{\sqrt[3]{D_p}} \quad \text{(13)}$$

But since pump and motor are to have equal displacements, we may say

$$N_m = \frac{.90 \times 2000}{\sqrt[3]{D_m}} \quad \text{(14)}$$

From Equation 5 and the assumption of $y_{CL} = .95$ we have

$$T_m = \frac{D_m \times .95 \times 1000}{2\pi} \quad \text{(15)}$$

Expression 12 is now

$$10 = \frac{2\pi}{12 \times 33{,}000} \times \frac{D_m \times .95 \times 1000}{2\pi} \times \frac{.90 \times 2000}{\sqrt[3]{D_m}} \quad \text{(16)}$$

From which we obtain $D_m = D_p = 3.54$ cubic inches per revolution.

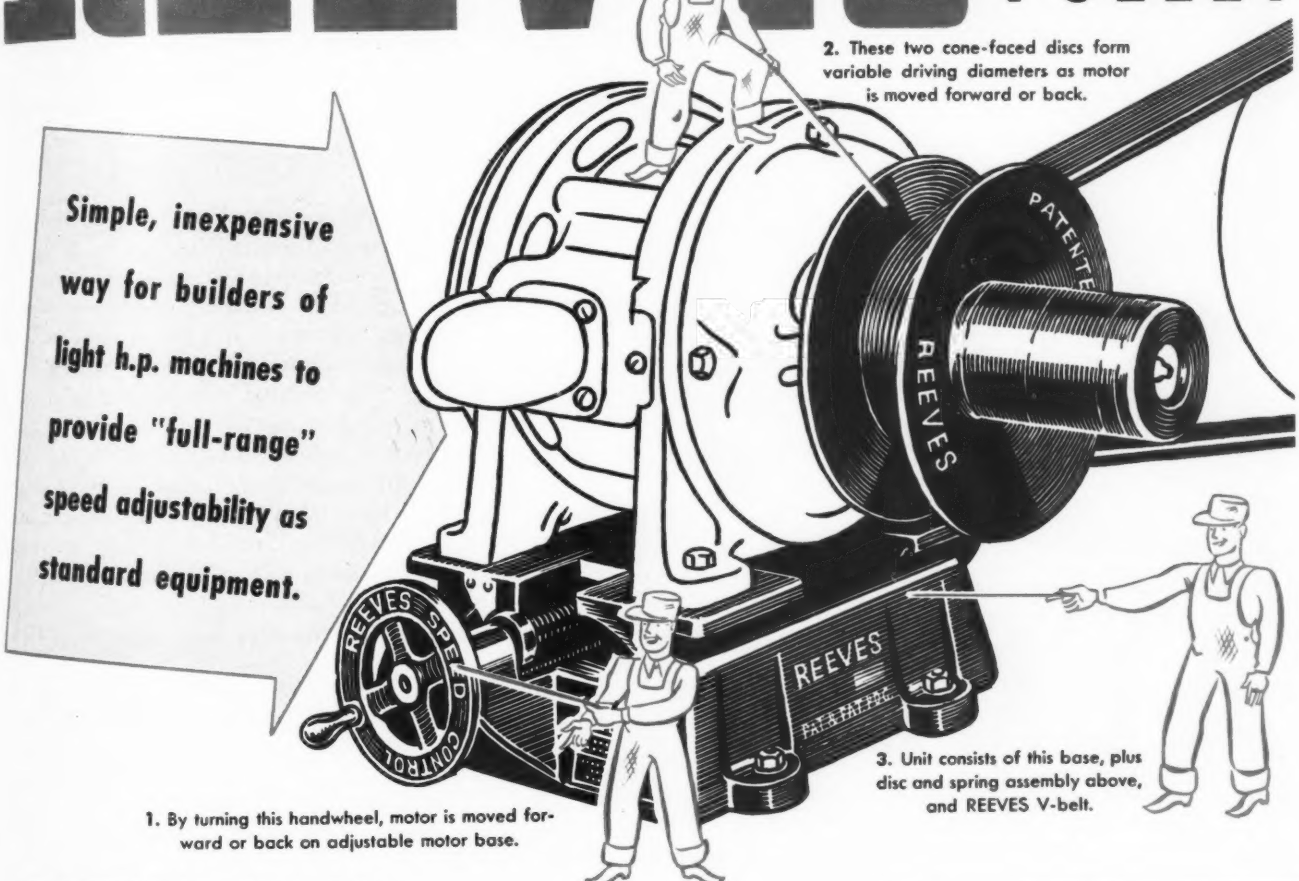
6. The speed at which the pump should operate should therefore be approximately,

$$N_p = \frac{2000}{\sqrt[3]{3.54}} = 1310\, RPM \quad \text{(17)}$$

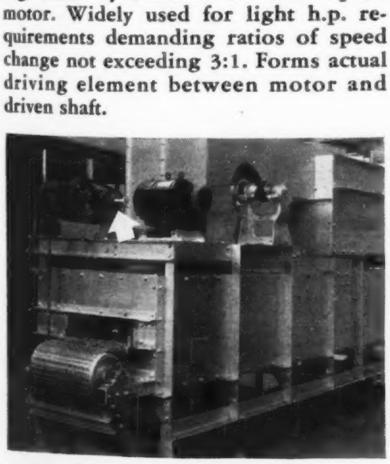
This is not a standard electric motor driving speed. To obtain a practical result it is necessary to choose the next larger displacement pump and motor commercially available, and operate at the nearest lower standard induction motor speed, which in this case is 1150 revolutions per minute.

It must be emphasized that the treatment here given of the factors involved in the design of a variable speed hydraulic drive is by no means complete, inasmuch as there are many minor but nevertheless important details of design not mentioned. For example, the selection of pipe walls for strength, leakage replenishment equipment, safety relief valve considerations, oil cooling and filtering, etc., have not been considered. Furthermore, the data proposed above for calculation of the principal transmission component sizes is offered as a fair estimate of average conditions. Results obtained should be treated as good approximations, to be checked by specific performance data applying to the hydraulic equipment chosen for use. Such information is readily obtainable from the engineering organizations of hydraulic manufacturers from whom is also available detailed engineering advice concerning the minor auxiliary components not mentioned here, but necessary for successful performance of the circuit.

REEVES Vari-Speed MOTOR PULLEY



● The REEVES Vari-Speed Motor Pulley is especially designed for direct mounting with any standard constant speed motor. Widely used for light h.p. requirements demanding ratios of speed change not exceeding 3:1. Forms actual driving element between motor and driven shaft.

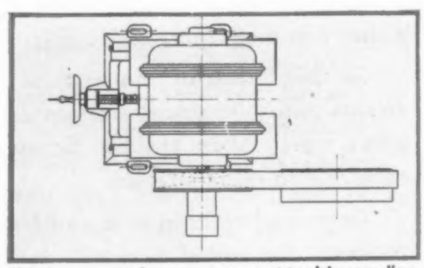


Heat treating furnace, built by Industrial Heating Equipment Co., equipped with REEVES Vari-Speed Motor Pulley.

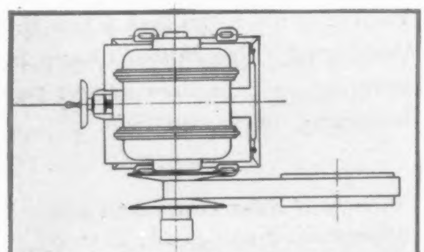
Two cone-faced discs are mounted on the standard motor shaft extension. Motor is mounted on adjustable base. Through handwheel control, motor is moved forward or back, varying depth of throat between discs. A REEVES V-belt, running between discs and connected to driven machinery, is kept under constant tension by a compression spring. As depth of throat is varied, V-belt assumes different diameters of contact against discs.

Speed adjustments are infinite between the limits, and are made while equipment is running. Speed settings accurately maintained. Constant torque; variable horsepower. Sizes: fractional to 15 h.p. Countershaft for unusual speed reduction or increase; also type for mounting gear reducer. Send for Catalog G-397, giving complete information on REEVES line.

REEVES PULLEY CO., *Dept. H,* **Columbus, Ind.**



Maximum speed—motor nearest to driven pulley



Minimum speed—motor farthest from driven pulley

REEVES ACCURATE POSITIVE SPEED CONTROL

Aluminum Bronze Fills Design Need

(*Continued from Page* 51)

moved by abrasion, machining, or other means, it immediately re-forms.

To date perhaps the widest use for aluminum bronze in this field lies in pickling equipment installations in which the corrosive medium is usually 5 to 7 per cent dilute sulphuric acid at temperatures just under boiling. Crates, baskets, chains and steam injectors of the alloy are in wide and successful use.

In the petroleum refining industry, aluminum bronze does an outstanding job in the handling of acid sludge, as barometric condensers, and as pumps and pump parts handling sea water, ethyl gasoline, and certain other specific corrosives encountered regionally or in individual refineries.

One warning in connection with the use of aluminum bronze or, for that matter, any bronze for corrosion resistance is in order. This is the occasional appearance of what is usually called "electrolysis," or galvanic action set up between unlike alloys in an acid solution. In such cases preferential corrosion may take place and cause rapid failure.

Solutions Not Recommended

Mention has been made of the fact that aluminum bronzes are not universally corrosion resistant. It is a well-known fact that no one commercial alloy will withstand all corrosive agents, and the following are some of the ones for which aluminum bronze in particular is not recommended:

Nitric acid	Copper sulphate
Mixed acids	Ferric compounds
Ammonia	Mercuric compounds
Ammonium hydroxide	Zinc sulphate
Ammonium sulphite	Chlorine compounds

A few of the many corrosive media in which aluminum bronzes have proved outstandingly successful are:

Acetic acid	Fatty acids
Acid sludge (refinery)	Hydrochloric acid
Beer	Hydrofluoric acid
Benzene	Hydrogen sulphide
Brine	Lactic acid
Carbon tetrachloride	Sulphuric acid
Citric acid	Tannic acid

The three general fields of service, notably bearings, gears, and corrosive applications, listed to this point are those into which the greatest number of universal applications may be grouped. In addition there are a great many other types of service for which the aluminum bronzes are suited by virtue of certain physical characteristics not mentioned previously.

The resistance welding field, for example, makes wide use of the alloy for two reasons. First, it has greater resistance to the adhesion of weld spatter

than any other copper-base alloy. Welding dies or jaws made from aluminum bronze are readily kept clean. Secondly, its electrical conductivity (10 to 20 per cent) is sufficient to permit of its being used as contacts in butt and flash welding where there is ample surface in contact with the work for passage of the necessary current. Other alloys of the beryllium-copper, copper-chrome, and copper-beryllium-cobalt series are better suited where contact area is small, and higher electrical conductivity values are required.

A special grade of aluminum bronze is widely used for forming and drawing dies. It has been found that this alloy largely eliminates scoring and galling, yet is hard enough and has sufficient compressive strength to withstand the service imposed by forming, drawing, and stamping operations.

Aluminum bronzes are also noted for their ability to stand up under high temperature service. In no case are they to be classed with heat-resistant steels, yet several instances (which involve low unit stresses) are on record wherein they have operated successfully at temperatures at or near 1000 degrees Fahr.

By way of brief resume it may be pointed out that there exist two general classifications of aluminum bronze: The low-iron type which contains less than 1.25 per cent iron, as an alloy addition, and the high-iron type which usually contains from 2.5 to 4 per cent iron. This latter type usually has smaller grain size, greater hardness and higher resistance to wear.

Aluminum Controls Properties

Of even greater influence on the properties of the material is aluminum content. Less than 9 per cent results in an alloy of predominantly "alpha" crystals, which are soft (115 brinell) and very ductile. An alloy containing from 9.5 to 11.8 per cent aluminum will have a structure partially or mainly, depending on cooling rate, composed of "beta" or "eutectoid" phases which are much harder (240 to 270 brinell) and quite brittle. Aluminum bronze containing over 11.8 per cent aluminum will contain a percentage of "delta" crystals which have a brinell hardness of about 460, and are, of course extremely brittle. By varying the proportions and forms of these various phases through aluminum content and heat treatment, an alloy can be obtained which is "tailor-made" for the application at hand. Table I lists a few of the more common alloys thus produced.

Aluminum bronze is not only capable of being heat treated for physical property improvement, but is also subject to the reverse effects. That is, castings of heavy section which cool very slowly in sand may fall far short of reaching specified physicals. In the case of a slow-cooled alloy of over 10 per cent aluminum content, elongation may drop from 15 per cent to 3-5 per cent, and tensile strength from 75,000 pounds per square inch to 60,000 pounds per square inch. At the same time hardness will increase and yield strength will approach tensile.

Theory of Elasticity in Practical Design

(*Continued from Page* 63)

are determined as

$$S_1 = -2B(\beta - \sin\beta)$$

$$S_2 = -2B(\beta + \sin\beta)$$

Now by satisfaction of the boundary condition the constant B may be determined. Along the unloaded portion of the plate $\beta = 0$ and $S_1 = S_2 = 0$, giving a boundary free of stress. Underneath the load $\beta = \pi$, $\sin\beta = 0$, $S_1 = S_2 = -2B\pi$, and the shear stress is zero in all directions. Since S_2 bisects the angle β, it is perpendicular to the boundary and equal to $-q$. Equating and solving for B

$$B = \frac{q}{2\pi}$$

Substituting in the equations for S_1 and S_2, the principal stresses of the combined system are finally

$$S_1 = -\frac{q}{\pi}(\beta - \sin\beta) \qquad (112)$$

$$S_2 = -\frac{q}{\pi}(\beta + \sin\beta) \qquad (113)$$

Angle β may vary from 0 to π. Since β will always be greater than $\sin\beta$, except when both are equal to 0, S_1 as well as S_2 will always be compression. As the distance of the stress point R from the load is increased without limit β approaches O and the stress vanishes, giving zero stress on the "infinite boundaries" of the plate. A circle has been drawn in *Fig.* 57 through the end points O' and O'' of the load, and through the stress point R. For all points on this circle, β is constant and therefore the stress is constant.

Stress Difference Is Constant along Circle

The stress difference Q is given by

$$Q = S_1 - S_2 = \frac{2q}{\pi}\sin\beta \qquad (114)$$

Stress difference Q will also be constant along the circle, the photoelastic pattern consisting of a series of circular rings somewhat as the pattern for a concentrated load except that the rings will pass through the load end points. Also, they will build up to a maximum value at $\beta = \pi/2$ and then reduce to zero at the load, where the stress condition is a hydrostatic head or point of zero principal stress difference.

Along the perpendicular bisector of the load, S_2

(*Continued on Page* 128)

(*Continued from Page* 122)

is perpendicular to the boundary. The maximum Q is at $w/2$ below the boundary and the center of the stress circle is on the boundary, giving a semicircle. The fringe value at this point is

$$n = \frac{Qb}{H} = \frac{2qb}{\pi H} = \frac{2W}{\pi H w} \quad \ldots\ldots\ldots (115)$$

$$\text{for} \qquad \beta = \frac{\pi}{2}$$

where W is the total load $= qwb$.

In *Fig.* 61 is a photoelastic picture of a uniform, partially distributed load acting on the boundary of a 7¼ inch square plate. $b = .272$-inch and $H = 87$. Load of 359 pounds is applied over a width of .500-inch. The characteristic rings may be plainly seen, similarly as for the concentrated load of *Fig.* 50. The semicircular fringe at the level 6 is slightly open at its ends due to the load not being uniform. This also accounts for the ripple effect immediately under the load. Since the fringe spacing for this loading is large, a light field photo was taken. In this type of picture the axis of the quarter-wave plates are set parallel and at 45 degrees to the polarizer, giving maximum brightness. The interference fringes are then at half steps. The use of this in conjunction with *Fig.* 61 doubles the fringes†. To bring this point out these two pictures have been superimposed in *Fig.* 62. Registry is slightly out laterally. As can be seen, the effect on an open pattern is as if the model sensitivity were doubled, i.e., H were halved.

Fringes Are Close to Theoretical

In *Fig.* 63 the Q stress along the center line of the plate has been plotted in terms of w, the load width. Values taken from the photoelastic pictures, *Figs.* 61 and 62, are plotted in the dotted curve. Agreement in this case is not as good as would be anticipated, indicating a possible small error in the experimental work. It should be noted, as previously mentioned, that the load is not quite uniformly distributed. Also the ratio of plate size to number 2 fringe diameter is only about 4. The maximum stress by theory is 11½ per cent less than the photoelastic.

The characteristic ring fringes, from the uniform loads or from the concentrated loading previously discussed, may be seen in many of the photoelastic pictures that have already been given, even in many of the shapes that are far from the theoretical semi-infinite plate. In the photoelastic series‡, the outer fringes of *Fig.* 18 come surprisingly close to a cirlar shape and to passing through the load-end points, except as disturbed in the lower half by the "crotch." As the maximum fringe point is passed, approaching the boundary, the pattern breaks down into two separate groups at each end. This is due

† This method was suggested by R. Weller of State College of Washington in a letter to the writer.
‡ MACHINE DESIGN—March to July, 1940.

to the concentration of load at these ends, simulating two concentrated loads due to the curvature of the model contour, making the ends of the loading blocks "bear-in" heavier than the mid-portion. Knowledge of the approximate pattern to be expected at this point, from the analysis of the semi-infinite plate, furnishes recognition of the lack of load uniformity at a glance.

It is interesting to note that the maximum shear (equal to $Q/2$) is zero at all points along the boundary even under the load. It increases to a maximum of q/π at one half the load length below the surface and then falls away. Further discussion of the significance of this is reserved for a later article.

The preceding discussion demonstrates how the general photoelastic pattern in the vicinity of a loaded body can be predicted and how some idea of the stress distribution may be gained in pieces of finite shape from the analysis of the semi-infinite plate. The next article will discuss theoretically concentrated loads on rollers and wedges. This will be followed with an article analyzing the actual load distribution at "concentrated" contacts of bodies, the Hertz equations so much used in the design of rollers, cams and other contacting bodies.

Meetings and Expositions

Aug. 25-29—

National Association of Power Engineers Inc. National Power Show and Mechanical exposition, and fortieth annual meeting will be held at the Fifth Regimental Armory, Baltimore. Fred W. Raven, Room 1717, 176 West Adams street, Chicago, is secretary.

Sept. 8-12—

American Chemical society. Semiannual meeting to be held at the Chalfonte Haddon hall, Atlantic City, N. J. Dr. Charles I. Parsons, Mills building, Washington, is secretary.

Sept. 23-26—

Association of Iron and Steel Engineers. Annual meeting to be held at Statler hotel, Cleveland. Brent Wiley, 1010 Empire building, Pittsburgh, is managing director.

Sept. 28-Oct. 2—

American Mining congress. Annual metal mining convention and exposition to be held at Fairmont hotel, San Francisco. Julian D. Conover, 309 Munsey building, Washington, is secretary.

Oct. 6-10—

National Restaurant association. Annual meeting and exhibition to be held at the National Restaurant Mart, Chicago. Frank J. Wiffler, 666 Lake Shore drive, Chicago, is secretary.

Oct. 13-14—

Steel Founders' Society of America. Fall meeting to be held at The Homestead, Hot Springs, Va. Raymond L. Collier, 920 Midland building, Cleveland, is secretary.

Oct. 19-24—

American Welding society. Annual meeting and exhibition to be held at the Bellevue-Stratford hotel, Philadelphia. Miss M. M. Kelly, 33 West Thirty-ninth street, New York, is secretary.

Oct. 20-24—

American Society for Metals. Annual meeting and exposition to be held at the Benjamin Franklin hotel, Philadelphia. W. H. Eisenman, 7301 Euclid avenue, Cleveland, is secretary.

Oct. 20-25—

Dairy Industries Supply Association Inc. Exposition and meetings are to be held in Toronto, Canada. Roberts Everett, 232 Madison avenue, New York, is executive vice president.

Nov. 12—

American Die Casting institute. Annual meeting to be held at the Cleveland hotel, Cleveland. Kenneth C. Castleman, 420 Lexington avenue, New York, is secretary.

Business and Sales Briefs

PLANS are being made by the Torrington Mfg. Co. for the erection of a new one-story building, to be attached to the present plant at Torrington, Conn. The new floor space will add to facilities available for manufacture of propeller fan blades and blower wheels. Occupancy is scheduled tentatively for October.

♦

Floor space of the Elastic Stop Nut Corp., Union, N. J., has been doubled to meet increased demand for its line of self-locking nuts.

♦

Joining the Roller-Smith Co. as sales manager in 1939, W. R. Swoish has been appointed vice president in charge of sales.

♦

Jessop Steel Co., Washington, Pa., has appointed H. E. Doughty manager of its recently established Philadelphia branch, located at 225 South Fifteenth street, Philadelphia.

♦

To take care of the growing demand for time control instruments, The Paragon Electric Co. has moved its manufacturing facilities from Manitowoc to Two Rivers, Wis. In its new location the company will occupy a building which is double the area of its former plant. Executive headquarters will remain in Chicago at 37 Van Buren street.

♦

Eastern district manager of Ampco Metal Inc., Milwaukee, Ray J. Thompson, has changed his address to 57 White avenue, West Hartford, Conn. Mr. Thompson has supervision over offices at Springfield, Mass., New York, Philadelphia and Washington.

♦

Frederick Post Co. has announced that the John R. Cassell Co., New York, has taken over the sales of the company's line of drafting materials in the Metropolitan New York area. The Cassell company is located at 110 West Forty-second street, New York.

♦

Allis-Chalmers Mfg. Co.'s office in Toledo is now located in the Toledo Trust building. W. R. Mengel is manager.

♦

Winfield H. Smith Inc., Springville, N. Y., is celebrating its fortieth anniversary. As a producer of speed reducers, gears and power transmission machinery, the company was founded in 1901 in Buffalo and in 1925 was expanded and moved to Springville.

♦

Two new additions to buildings have been announced by Phelps Dodge Copper Products Corp., Elizabeth, N. J., one of which is a storage warehouse already in use.

♦

J. Frank Oehlhoffen, who has been serving as assistant sales manager of Bantam Bearings Corp., South Bend, Ind., has been advanced to sales manager.

♦

W. L. Schneider, previously sales manager of The Falk Corp., Milwaukee, has been promoted to the position of vice president of sales. He will also

continue in the supervisory capacity of director of sales. T. F. Scannell succeeds Mr. Schneider as sales manager and will be responsible for active sales of the company's line. He had formerly been assistant to the sales manager. J. B. Kelley, in recent years in charge of the company's flexible coupling business, has been made assistant sales manager.

♦

Twenty-fifth anniversary of the founding of Logansport Machine Inc., Logansport, Ind., is being celebrated, and special tribute being paid to Frank B. Wilkinson. The company began from a modest beginning in July 1916 and has grown considerably, making a name for itself in the air and hydraulic equipment fields.

♦

Recently the Monsanto Chemical Co., St. Louis, opened a new plant in Springfield, Mass., for production of its Resinox plastic molding materials.

♦

To meet the rapid increase in industrial activity in the Cincinnati Territory, R. D. Yoder has been transferred from the Milwaukee office of Cutler-Hammer Inc., to Cincinnati.

♦

J. Arthur Deakin has been appointed Eastern district manager for McKenna Metals Co., Latrobe, Pa., with headquarters at New York.

♦

Appointment of Donald S. McKenzie as sales manager of the plastics department of the General Electric Co., has been announced. He will be located at the department's headquarters at Pittsfield, Mass., succeeding W. H. Milton Jr., recently named assistant manager of the department.

♦

Succeeded by Charles C. Limbocker as president of Wolverine Tube Co., Detroit, H. J. Hooks has resigned as president due to ill health. Mr. Limbocker will continue as chairman of the board.

♦

Connected with the motorized speed reducer sales for the past four years, Max L. Robinson is now sales manager for Janette Mfg. Co. He was formerly sales manager for the Roth Bros. & Co. division of Century Electric Co.

♦

To expand its production facilities for the manufacture of glass fibers for electrical insulation, Owens-Corning Fiberglas Corp. has purchased a plant in Ashton, R. I.

♦

Due to an expanded production schedule, Colts Patent Fire Arms Mfg. Co. has moved its electrical division into larger manufacturing quarters in Hartford, Conn.

♦

Construction of a second unit of its new plastic plant at Wallingford, Conn., has been started by the American Cyanamid Co. The plant will make several kinds of synthetic resins. The first unit is expected to start operations this summer.

♦

Offices of the Aircraft Screw Products Co. have been moved to 47-23 Thirty-fifth street, Long Island City, N. Y.

♦

Correction: Offices of American Felt Co., 315 Fourth avenue, New York, were not moved to Glenville, Conn., as reported in the June issue of MACHINE DESIGN. The Glenville mill and the adjacent executive offices have been completed and in operation for some time. The New York office, however, will continue as a sales and distribution office.